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PREVENTING LOSSES
IN
FACTORY POWER PLANTS

BY

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To

E. H. M

INTRODUCTION

There are usually four stages in the industrial evolution of a productive civilization such as that of the United States. The first is the frontier stage. All the men and most of the women work with their own hands. The important industries are those which have to do with the needs for food and clothes and housing. There is little surplus wealth and little spending. The hunter and the trapper is gradually being replaced by the farmer and the cattleman. The factories are small, and supply a local market: the employed class is not numerous at any one place. Each village or town is largely self-dependent. There are many mechanics in business for themselves.

The second era appears with the miner, the productive manufacturer and the development of transportation. The industries produce more than the local market can use, and more of a different kind, by reason of the aggregated labor and its economies. Supported as to food and clothes and housing, and with the highway and the railed way seeking to convey goods and raw material, an excess of wealth beyond that required to meet primary needs seeks investment and return. Greater factories arise, coordination of competitive interests takes place, and the era of "big business," of extending railways, of good highways, of luxury, leisure, of culture and of art begins.

The third stage may be called the era of Refinement in Production. These are the days of better business management, of economies of administra-

tion, of lowered costs of production and of power, of hydro-electric developments, of co-operation in production, of effort to reduce the cost of living.

The fourth stage is the era of Uplift. The captains of industry see not alone that it is their duty to make the producer wealthy, and the factory efficient and economical, but also that the human element must be healthy and life worth living. These are the times of securing safety and sanitation, of a recognition of the truth that the life is more than food and the body than raiment; that the employer is his brother's keeper.

Now the book of which these few lines are an introduction belongs to the third epoch and the beginnings of the fourth. It concerns itself with the reduction of operating costs in the power-plant, and aims to apply the methods of sound science to the problems of the production of power. The Author has specialized in the field of boiler-plant economy and fuel, and writes with the authority of experience in this and related departments of research. The treatment of power-plant losses is the outcome of study of the usual ignorances of executives and superintendents. The application of the principles advocated has actually effected savings of many thousands of dollars annually. The treatment is to convince the business end of the productive process, as represented by the owner and manager. It should be convincing, as the combination of theory and practice has been experimentally worked out. The power-plant has not been as attractive a field as some others to the so-called efficiency engineer; and the Author has had the privilege of recognizing the human factor in its basal relations to success. This puts the book in its pioneer place of literature as respects the fourth epoch of industrial evolution, and the ninth chapter should receive careful study. Data and conclusions on the combustion process, fuels, testing of steam generators, and the reporting

on any investigations made are features of the later chapters.

The relations of the Author and the writer of this introduction were at first the pleasant ones of the professor of engineering and the enthusiastic student. These have continued and developed into those of the affection of the older man for the younger and interest in his successful achievements. The start of the younger in the lines of thinking that the mechanical engineer must be also a business man is the older man's contribution. The development of that idea is the younger man's achievement as this presents it.

FREDERICK REMSEN HUTTON,

Past President, American Society of Mechanical Engineers.

PREFACE

This book has been written expressly for the use of the owners and managers of our manufacturing industries.

My object is to set before them, in as brief and direct a manner as possible, the basic information which they require for the intelligent handling of their power-plant problems.

With this idea in view, I have in my introductory chapters laid stress upon those fundamental principles and natural laws which are essential to a proper understanding of the numerous questions of specific nature which are later more fully developed.

As a feature of consequent importance I have given the actual application of the theories to the definite problems in hand, in order to show their direct relation to the striking economies which result from their practice.

All of these matters are broadly covered in the first nine chapters, and are so composed as to present a bird's-eye view of the entire subject for easy review by those executives whose limited time would preclude a more exhaustive study. For such, this first section alone has been specifically prepared.

The remainder of the book is given over to the further and more detailed study of the questions naturally arising from a perusal of the first part, and logically belongs more especially in the field of the mechanical superintendent and the factory engineer.

The book, as a whole, and, in so far as its limited size allows, represents present practice in the work of improving factory power-plant efficiency, and indicates the trend of advance which the future portends, both on the side of equipment and of operation.

Were I permitted the publication of but a single chapter of this book my immediate choice would be in favor of the ninth on the "Human Factor". For, by confining my expression to this single chapter, the object of the book would suffer in a less degree than would be the case if I had to be contented with *any six of the others.*

For the subject-matter of this work my principal acknowledgments are respectfully offered to my clients, the manufacturers, in whose establishments and with whose co-operation I have been enabled to study and apply the principles of efficiency for the building up of economy and the reduction of preventable losses in factory power plants.

David Moffat Myers

New York, October, 1914.

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PREVENTABLE LOSSES IN FACTORY POWER-PLANTS

Chapter I

THE DETERMINATION OF EXISTING LOSSES

EVERYONE who is any way connected with the great problems of the times is vitally concerned with the extensive subject of preventable losses. Broadly as well as definitely considered, all the questions of today depend for their ultimate solution upon a thorough and exhaustive analysis of preventable losses. The science of medicine, the practice of surgery, are aimed at prevention beforehand rather than an attempted cure after the physical breakdown has occurred. This results in the saving of life and health, the improvement of human effort,

and the countless benefits that are produced in so many directions. When preventable losses have been eliminated, increased efficiency is the direct result.

This simile may be extended indefinitely. Take the great social problems. The workers and reformers are striving diligently to understand those elements in our social system which are at the foundation of the existing losses, mental, moral and physical, toward the prevention of which they have devoted their efforts. The aim is first by study definitely to locate, and then finally to eliminate those elements which destroy human efficiency. No matter what the problem, the method of attack both logically and actually is the same. Determine the preventable losses, gain knowledge of their respective causes, and then intelligently and effectually eliminate them.

Turning now to a consideration of problems of engineering, we begin to deal with elements that are more definite. We have a vast amount of reliable data from which formulæ have been deduced. These formulæ have been applied to actual work and the work has proved their value.

The new and growing science of efficiency engineering as applied to production, although as yet in the days of its infancy, has

achieved some very remarkable results. The various leaders and workers in this field, although differing considerably in their methods and in their theories, are all united in the one effort—that of reducing preventable losses to a minimum.

But in this field the engineer is compelled to make for his work in hand a definite or maximum standard of efficiency, and toward this standard he directs his efforts. It follows that the careful establishment of this standard is perhaps the most important feature of his work. This standard or degree of efficiency, when set by the competent expert, is always a practical one, and can be attained by the best known practice of the profession with savings which at times seem wonderful when compared with old-time practice.

For all work of high accomplishment an ideal must be had toward which to work. This ideal must be kept constantly in mind and all results judged in comparison with this work, which should represent 100 per cent efficiency. The physician, the social worker, the efficiency engineer, must each possess a vision of their respective ideals, their marks of 100 per cent efficiency, the physically perfect body and mind, the perfect unit of human society, the perfect process

for production without loss. All imaginary, all visionary, but absolutely essential and with tremendous power for inspiration and attainment.

And here, too, arises great difficulty. For in most fields of operation there is wide divergence of opinion as to what constitutes the ideal, the perfect result. In medicine great discoveries are made. An additional disease is brought under control. But the question is how much *more* can medicine accomplish even if perfection of knowledge is reached? No answer is possible.

In society great reforms have been effected. Great good has resulted. But what is the ultimate good that will result, the 100 per cent moral efficiency? What is the standard, the perfect human ideal whereby all lives may be accurately and truly measured? It is the very diversity of ideals, the lack of knowledge of the one perfect universal truth, that prevents more rapid progress toward that perfect unit and body of human society so desired and so necessary.

In production or efficiency engineering, after great savings have been made, more savings still are discovered to be possible. It is impossible to know at what point the limit of economy will be reached. It cannot be known what constitutes 100 per cent effi-

ciency or how much preventable loss still remains after all brain power and training have been exhausted. The same truth applies to the science of machine design. A device is invented which increases production and reduces the unit cost perhaps 25 per cent. Can a still better machine be devised which will reduce to a still further extent losses that in the light of present knowledge and experience evade all attempts at discovery? The answer is a question. *There is no 100 per cent efficiency with which to judge, and future improvement is entirely problematic both as to its possibility and as to its extent.*

Among all the problems of our time there is a very important one which possesses the tremendous advantage of a definite determinable ideal. The 100 per cent efficiency of the perfect power plant is known and understood. The processes involving the production of ''power'', in glowing contrast to almost all other processes, are subject to absolute and ultimate analysis. This statement is true for all sources of power and for all methods of converting latent energy into active work.

As an instance consider water power. If we have a stream supplying 1,000 pounds of water per minute with an available head of 33 feet, a perfect turbine or water wheel

would convert all the potential energy of the water into useful work; that is, we should receive 33,000 foot pounds of energy per minute because that is the full amount of energy in the water. It is therefore impossible to obtain more than this amount. Actually we shall obtain much less, and with certain forms of wheel perhaps only one-half this amount. In this case the efficiency of our process is exactly 50 per cent and our losses are 50 per cent. We know that our limit of possible improvement in our machine is 100 per cent, or a maximum of 33,000 foot pounds per minute or one horse power. Hence we have *fixed for us by natural laws* a definite standard, a 100 per cent efficiency mark which represents absolute perfection. If a turbine could be designed which would develop energy at the rate of one horse power from 1,000 pounds of water falling 33 feet per minute we should know definitely that we had evolved a perfect process and that no more improvement would be possible. There would be no further losses to overcome. Owing, however, to friction and other fixed causes we do not expect to obtain 100 per cent efficiency. If this could be done our turbine would be able to pump the water it had used up over the dam again, thus reestablishing its source of power and we

should then have the much dreamed of "perpetual motion".

I have more than once been called upon by capitalists to investigate perpetual-motion machines, usually called by other names for the purpose probably of misleading the innocent investor, and I will offer to such an added word of advice. If all friction and other natural inherent losses could actually be overcome and a perpetual-motion machine could be designed and made to operate, it would still be absolutely worthless for the purpose of running machinery or doing any kind of work whatever. This, for the simple and comprehensive reason that it would have to expend every foot pound of energy developed in order to re-establish its source of power, and there would therefore be no surplus energy available for purposes of outside work. If it could be conceived that a perpetual-motion machine could be made to work, it would come to an instantaneous standstill as soon as a dynamo or any other load was imposed upon it.

This simple example illustrates accurately and with forcible truth the fundamental principle underlying and making possible all work of genuine investigation and improvement in the field of power-plant operation and design. This underlying truth is known as the

principle of "the conservation of energy." Otherwise stated, energy can neither be destroyed nor increased. It is true that energy can be transformed from one manifestation to another, and this is a most common occurrence in Nature. Thus the heat of the sun evaporates water from the sea. This water is precipitated in rain on the mountains where vast reservoirs of latent energy are available by application of the water wheel, and mechanical energy is thus the direct result of the evaporative power of the heat rays from the sun.

In power work the mechanical energy of the water wheel is converted into the electrical form of energy through the medium of the dynamo. The latent heat of coal through the agencies of the furnace, the boiler, and the engine, is transformed into mechanical energy. Through the generator connected to the engine it becomes electrical energy, and once again through the motors it is converted back into the mechanical form of work on line shaft and moving machinery. Thus we have many instances we may call to mind which illustrate the transformation of one form of energy to energy of another form.

In the foregoing example of transformation it will be realized that the final amount

of energy developed in the machinery is less than the energy supplied to the motor which drives it. That supplied to the motor by the dynamo is less than that it in turn received from the engine, while the engine transforms into useful work only a very small fraction of the heat energy it derived from the boiler, and finally the boiler gives to the engine in the form of steam only a part of the heat that was applied to the boiler in the form of fuel. Thus we have a series of energy losses, at least one for each transformation, all the way from the coal in the boiler room down to the machinery in the factory which it is intended to operate.

This being the case, the reader may object that the conservation principle does not hold true, for we have started with 100 per cent of heat energy in the coal and when we measure the amount of actual output at the dynamo or the motor we find only a small fraction of the original input, and the balance is simply lost or dissipated in some unaccountable way. The whole matter appears to be a mystery or else the principle of conservation is all wrong and unreliable.

As a matter of fact, however, there is no more beautiful method of proving the conservation theory than to start with 100 heat units of energy in the coal pile and to follow

each successive step in its various processes of transformation and transmission until we find at the end of this journey the exact amount of useful work our calculations will have called for.

The greatest feature of power investigation lies in our ability to account accurately for all units of energy, from their original liberation down to the final amount of useful work that has been produced. This final amount of energy or the useful output of the plant when compared to the original input gives us the over-all efficiency of the process. Thus if we find that by using a certain combination of coal, furnace, boiler, and engine we produce an amount of energy at the belt wheel equivalent to 2,545 heat units, and if we have burned 4 pounds of coal of 13,000 British thermal units per pound to do this, our input was 4 by 13,000=52,000 British thermal units. Our over-all efficiency in this case is then 2,545÷52,000=4.9 per cent, and our losses of transformation from heat energy to mechanical energy have been 95.1 per cent of the original heat in the fuel. This is a tremendous discrepancy between input and output, as well as quite an ordinary one; and were it not for our knowledge of the principle of the conservation of energy we should be able only to grope blindly

toward improvement. We would have no means of knowing in the first place whether any improvement were possible, and secondly we could not know in what direction to search for preventable losses.

But fortunately such is not the case. We are, on the contrary, able to analyze definitely and with satisfaction the entire energy loss and to account for each item, and furthermore to determine what part of the total loss is preventable.

In other words, we can by careful investigation and proper tests analyze each step of the process of power generation in such a way as to strike a balance with the original input of energy with all items accounted for. Then by concentrating our attention upon each individual item successively, we are able to determine which of them constitute preventable losses, whereupon they may be reclaimed by proper changes in operation or equipment and the over-all efficiency of the plant will thereby be increased.

In the days of Watt the power problem was one of entire mystery, and improvements came slowly by purely experimental hit-and-miss methods. The gradual determination and gradually increasing application of the principles of conservation of energy have changed all of this. In 1842 or thereabouts

Joule standardized the mechanical equivalent of heat. He found that 772 foot pounds of mechanical energy would raise the temperature of one pound of pure water one degree Fahrenheit, which is the unit of heat known as the British thermal unit. Later experimenters determined this mechanical equivalent more accurately to be 778 foot pounds.

By this "equivalent" we know exactly how much loss takes place when we burn so many pounds of coal and produce so many foot pounds or horse-power hours of work at our engine fly-wheel, for the heat energy of the coal is converted by its equivalent to the number of foot pounds of energy that would be produced at the engine fly-wheel providing the process were a perfect one. Then by comparing the horse-power hours or foot pounds the engine actually does produce with the perfect result, we have our percentage of efficiency and our percentage of loss.

When we turn to electrical energy our calculations just as readily and just as accurately translate the scales of our values either into those of heat or of mechanical work. Thus 1 kilowatt-hour equals 1.34 horse-power hours which equals 2,654,200 foot pounds or 3,412 heat units.

Stated in terms of power, which means rate

of work, we have:—1 kilowatt = 1.34 horse power = 3,412 heat units per hour.

Thus it is seen how readily and accurately the entire process of power generation can be followed through each successive step from the heat units in the coal pile to the final amount of mechanical or electrical energy which has been the ultimate object of the various changes set in motion.

I have made use of the term "losses" to represent that part of the original energy which does not appear at the end of the process as available or useful work for the purpose in hand. In other words, I have spoken of as "loss" the difference between the input and the output in any power process—that is, the discrepancy between 100 per cent efficiency and the actual efficiency obtained. In the absolute, however, this discrepancy or apparent loss is only that part of the original energy which has been absorbed by results other than the one desired. A simple illustration may be of value. Suppose an engine driving the machinery in a shop is developing 100 horse power measured at its fly-wheel and we find that only 50 horse power is reaching the actual machinery. Then the efficiency of our transmission is only 50 per cent and we have a 50 per cent loss which is absorbed in friction, a natural cause

capable of absorbing unlimited amounts of energy. A part of this loss will probably be found to be preventable, say to the extent of one-half. Then by reducing the friction to this extent by known means of transmission improvement, we shall have increased our efficiency from 50 per cent to 75 per cent. There will yet be a loss of 25 per cent, but we shall have gained 25 ÷ 50 or 50 per cent of available machine power, or have reduced the required engine power by 33 1/3 per cent for the same machine power.

It is now seen that while there are no losses in nature, at the same time we do find misdirected and misspent energy prevailing when we desire to utilize all of it for the particular purpose in hand. Also it is true that in nearly all cases a part of the wasted energy can be recovered and applied to useful work. The term "waste energy" is the best to apply to what we have termed "loss," and helps us to remember the meaning and value of the principle of the conservation of energy.

In the practice of fuel economy in the boiler plant the first necessity is to determine the existing efficiency of operation. It is possible and readily practicable to determine what percentage of the heat in the coal is being converted into heat in the form of

steam in the boilers. The difference between the discovered efficiency and 100 per cent is the waste of heat (or energy) that is taking place. If, for instance, the efficiency is found to be 52½ per cent, then 47½ per cent of the heat of the coal is being wasted. An analysis of each item of this waste may develop the fact that 30 per cent of the total heat of the coal is escaping up the chimney. A further examination may prove that this is caused by the feeding of a large excess of air to the fire, and that 17½ per cent of this is unnecessary and can be stopped by using less grate surface and a better method of handling fires. In this case the 17½ per cent of waste heat is added to the old efficiency which increases it to 70 per cent. The saving in fuel at once becomes $17\frac{1}{2} \div 70$ or 25 per cent and the gain in steam for the same coal consumption is $17\frac{1}{2} \div 52\frac{1}{2} = 33\ 1/3$ per cent. It happens that this example is drawn from one of my recent plant investigations and that the improvement indicated was effected without the installation of any new equipment whatever. The saving is in constant daily operation and is merely typical of the application of the principle of conservation of energy to the boiler room department of power-plant practice.

Owners and managers of industrial plants

are frequently confused and often misled by the extravagant claims of promoters regarding the performance of their power contrivances. It is seldom that such claims are made in terms of true efficiency as this would at once discredit their statements. An instance of this kind was in connection with a smokeless boiler furnace, and purported test records were exhibited which showed an evaporation of as high as 17 pounds of water into steam (from and at 212 degrees) per pound of coal. With the best average coal obtainable (14,500 B.t.u. per pound) this would mean a thermal efficiency of 113.6 per cent. That is, they claimed not only the perfect absorption of all the heat the coal contained but over 13 per cent more. The principle of conservation gave timely warning, and shortly afterward I was called upon to conduct tests on this furnace which gave very ordinary results.

In summarizing the salient points of our introductory discussion we may select the following basic features as a foundation for subsequent research along the lines of preventable losses.

The determination and location of preventable losses offers the most natural and logical method of arriving ultimately at increased efficiency in every field of operation.

In nearly all branches of human endeavor there is no proved and universally accepted standard which represents ultimate perfection or 100 per cent efficiency. Consequently there is a lack of the necessary ideal. The result of perfect process is unknown. *Owing to this fact the existing degree of efficiency is impossible to determine. Therefore the percentage of loss or waste is also indeterminable, and improvements are made only as a result of more or less blind experimenting.* Furthermore, after an increase of efficiency *is* so obtained, there is no way of knowing how much further improvement is possible. This condition baffles research and makes progress entirely problematical both as to its possibility and as to its extent. All this trouble, this blindfolded groping, is due entirely to our inability to determine the perfect ideal, the mark of final attainment, the degree of 100 per cent efficiency.

In contra-distinction to nearly all other fields of research and progress, that of power generation and application is imbued by the laws of nature with a definite ideal of perfection beyond which we cannot attain, and toward which we may concentrate our efforts with the intelligence that our work is truly aimed toward the elimination of waste. This knowledge is based on the unassailable laws

of the conservation and the transmutation of energy. These laws together providing an accurate code of measurement of energy in its various forms enable the engineer to analyze with certainty all of the steps involved in the liberation, transformation, and utilization of energy as applied to the purposes of power-plant design and operation.

Just as the expert accountant is able to analyze the expenditure of one hundred dollars in a business enterprise and to show where some of them are wasted or misspent, and finally to strike a true balance between income and expenditure, just so truly and with as great a degree of accuracy a trained engineer may analyze and balance the expenditure of energy, from the original 100 per cent income or input to the final machine horse-power hours of useful work, and in so doing he may point out where certain portions of this energy are misspent or wasted and how they may be saved and converted into useful work.

There does not exist a power problem that is not capable of solution by the intelligent application of these principles of analysis. Furthermore, experience in this special practice has demonstrated that preventable losses have been found to exist in every power plant that has been investigated, and it is some-

what more than probable that they exist to greater or less extent in every power plant that was ever designed or operated.

While all types of plants are subject to the investigation and correction of preventable waste, I shall confine all subsequent sections of this treatise to the study of the betterment of efficiency in *factory* power plants. They not only comprise a very special field, owing to particular and varying requirements peculiar to them, but they have received less study and far less attention than the more completely developed central power-station common to public-service conditions. The factory plants are therefore capable of far greater improvement, and contain as a rule a very large percentage of preventable waste which is capable of being converted into a handsome profit. The truth of this statement will be demonstrated throughout this volume, not only by an explanation of the method of making investigations but also by a citation of typical savings that have been the direct result of this work when applied to numerous problems in factory power plants.

Chapter II

ATTAINABLE EFFICIENCY AND ORDINARY WASTES

THE local and practical use of the conservation principle has resolved the work of factory power-plant investigation into a definite and at the same time comprehensive code or system.

The object is to determine not alone the over-all efficiency of a plant, but to find out the individual efficiency, and conversely the waste, of each step in the generation and utilization of the energy under consideration. Bound up very closely with the practical aspect of the case is the matter of "commercial efficiency" which always demands the most careful investigation. For it is with this result that the business man is most vitally concerned. He desires to know how much he is wasting in terms of dollars and cents, and cares very little for this value expressed in heat units. He wants to be defi-

nitely informed how many dollars' worth of fuel he can dispense with, and not how much the combined thermal efficiency of his boilers and engines can be increased.

Commercial efficiency is thermal efficiency properly modified by monetary values which may be readily applied as demanded by any given case. The business man is not very much interested, though probably confused, by the statement that the internal-combustion engine develops a higher thermal efficiency than any other kind of prime mover, but he *is* interested in the question as to what kind of an engine will produce the greatest number of kilowatt-hours for one dollar when operating under *his local* conditions and special industrial requirements. Let us take an example of commercial efficiency, quite a common one. A boiler plant is equipped to burn high-grade soft coal of 14,500 B.t.u., costing $4 per short ton, at the high efficiency of say 75 per cent. (I have tested a well equipped plant doing exactly this work in every-day practice.) This operation will give 1,000 pounds of steam at a fuel cost of $0.1785.[1]

[1] Evaporation per pound of coal $= \frac{0.75 \times 14{,}500}{970.4} = 11.2$ pounds.

That is 75 per cent of the heat value of the coal (14,500 B.t.u.) is utilized, and a pound of water is evaporated into steam from and at 212 degrees for each 970.4 B.t.u. so utilized. This last figure is the unit of evaporation, and is the amount of heat required to gasify a pound of water under these

Now if there happens to be in the market a poorer grade of coal of 12,000 B.t.u., which it is known will develop under equally favorable conditions an efficiency of only 68 per cent but costing only $2.00 per ton, the fuel cost of evaporation may be made by its use $0.119 per 1,000 pounds of steam.[2] Therefore we may *reduce our thermal efficiency* and at the same time *increase our commercial efficiency* enough to gain a fuel-cost saving of

$$\left(\frac{0.1785—0.1190}{0.1785}\right) = 33\ 1/3 \text{ per cent}$$

If we know the two thermal efficiencies and their respective coal prices, we may determine a true comparison of commercial efficiencies by application of a very simple formula, retaining the present existing thermal efficiency as a basis. This formula is as follows:

Commercial Efficiency =

$$Ec \quad \frac{(B.t.u._2 \times Price_1)}{(B.t.u._1 \times Price_2)}$$

conditions without loss. The fuel cost of evaporating 1,000 pounds of water will be in this case $\frac{1,000}{11.2} \times \frac{\$4.00}{2000} = \$0.1785$.

[2] Evaporation $= \frac{0.68 \times 12,000}{970.4} =$ 8.4 pounds and fuel cost of 1,000 pounds steam $= \frac{1,000}{8.4} \times \frac{\$2.00}{2,000} = \$0.119$.

In which:—

Ec = the new thermal efficiency under consideration.
$B.t.u._1$ = the heat per pound of coal now burned.
$B.t.u._2$ = the heat per pound of coal proposed.
$Price_1$ = cost per ton of coal now burned.
$Price_2$ = cost per ton of coal proposed.

The commercial efficiency thus obtained is directly comparable to the original thermal efficiency of the boiler plant.

This emphasizes one of the practical applications of commercial efficiency. It will be noted, however, that a knowledge of thermal efficiency is at the foundation of the commercial value and is essential to it.

There are many other instances of the modification or translation of thermal efficiencies to meet financial requirements. One of the more important of these is the matter of determining the *net* financial returns connected with a proposed change. Thus under a given set of conditions it may be found that a certain device will save a definite percentage of the coal fired under the boilers. It is self-evident that if such a device is sufficiently expensive in first cost or in operation, the return on the investment will not warrant its installation. Also an accurate deduction must be made from the apparent saving to cover all fixed charges necessitated

by the proposed change. These charges must always include the items of interest, depreciation, repairs, added labor, and sometimes insurance, and the consideration of safety and reliability must never be neglected.

In other words, the true commercial saving to the plant owner is the annual gross saving of the device *less* all yearly charges of all kinds that would result from its purchase and use.

There have been cases in my experience where for instance an improved type of steam engine would reduce the consumption of steam and fuel, but where cheap fuel was burned so efficiently that at the consequent low cost of steam the saving in dollars per year would not have paid a good interest upon the cost of the change.

It is a fact, however, that in most factory plants savings can always be made which will pay for themselves, all charges included, within two or possibly three years, after which the savings are clear profit. In a great many cases the possible savings revealed by scientific investigation are so large that they will pay for themselves in one year and often in a few months.

"Power costs" are entirely a matter of commercial efficiency, remembering always that commercial efficiency is derived from

thermal efficiency properly modified with monetary values suited to any given set of local conditions. It may be said that the cost of power depends upon two factors, viz: —thermal efficiency and location; for the location always determines the cost and quality of fuel, of labor, and of equipment as well as the value of money.

I have said that the principle of the conservation of energy has enabled us to reduce the matter of power-cost investigation to a definite scientific system. This system, as just pointed out, has two factors, viz: thermal efficiency and monetary value, the two combined producing the ultimate commercial efficiency of the sought for result. I have no doubt sufficiently indicated the method of converting thermal efficiency into commercial efficiency, a purely arithmetical operation providing ample consideration is given to all entering conditions; and I will now pass on to an explanation and discussion of the underlying chain of thermal efficiencies which constitute the foundation of economy.

For the sake of clearness in this undertaking I will take an example of what may be considered a typical factory power plant.

During the present stage of our development steam power is employed in the great majority of factories. There are sound rea-

sons for this as well as for the prediction that steam will continue well into the future as the one greatest source of power for our industries. Therefore we shall select a steam plant for our illustration of factory power-plant analysis. The degrees of efficiency of the various parts of our plant will be selected as a fair average of the existing conditions of the times.

The boiler plant generally delivers a part of its steam to engines and pumps and the remainder to heating systems and process work. The exact distribution of this steam is determined by special tests made for the purpose, and as illustration I have selected a simple "Steam Balance" from one of my reports on a factory boiler plant.

STEAM BALANCE FOR BOILER ON 8-HOUR BASIS FROM 7 A. M. TO 3 P. M.

Average boiler horse power developed with full working load			222
Average boiler horse power used by main engine		131.0	
Average boiler horse power used by one feed and one vacuum pump		14.2	
Average boiler horse power used by first coat dry room		41.0	
Average boiler horse power to printing engines; Oil-pumping engine; Low-pressure boiler-feed pump	by diff.	35.8	
Total output		222.0	222

Engine efficiency

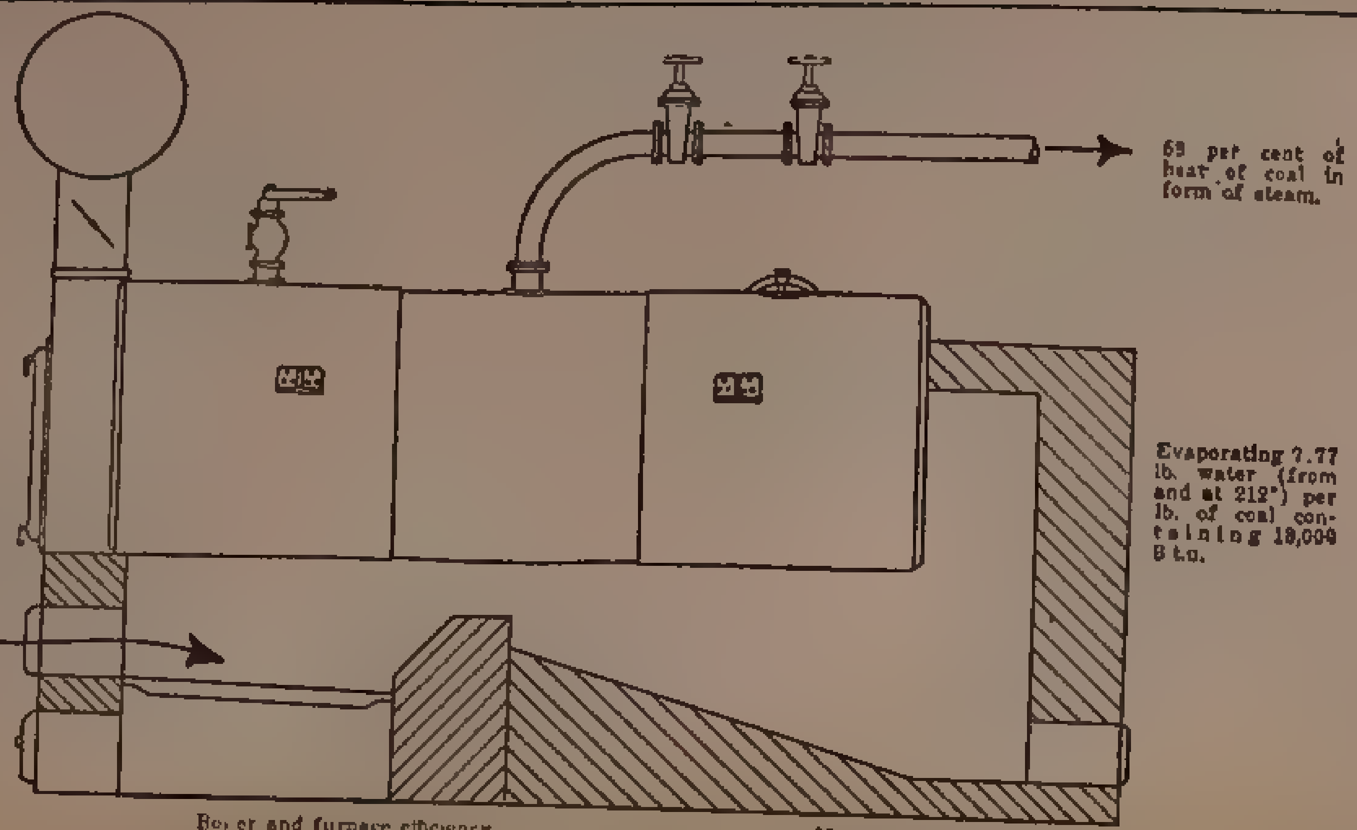
$$\text{Efficiency} = \frac{\text{out-put}}{\text{in-put}} = \frac{2,545}{34,830} = 7.3 \text{ per cent.}$$

3. ENGINE PLANT. RECEIVES FROM PIPING 57 PER CENT OF HEAT OF COAL. DELIVERS 4.16 PER CENT OF THIS IN MECHANICAL ENERGY AND DISCHARGES 52.8 PER CENT IN THE EXHAUST.

While these several uses of the steam are closely inter-related in their bearing upon our problem in general, it will facilitate analysis to treat these two methods of consumption in separate though parallel divisions. For the sake of clearness therefore the accompanying series of diagrams has been arranged to illustrate a plant in which all the steam from the boilers goes directly to the engines and no live steam is used for heating. (See insert facing this page.)

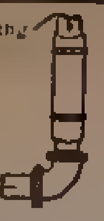
We have now seen a graphic illustration of the complete cycle and distribution of steam such as would be actually found in what may be truly said to be a typical factory power plant, providing all the steam from the boilers first entered the engines. Usually, however, a considerable portion of this original steam goes directly to heating and process work. In this case its cycles of utilization, waste, and return can be traced and measured in the same way that has just been illustrated in the case of the exhaust steam. This will be treated in the chapter on heating.

It will be noted that one item of our energy has not yet been followed through to the end, and that is the portion of heat energy which was converted into useful mechanical work by our steam engine. There



Boiler and furnace efficiency .... 68 per cent
Boiler and furnace waste .... 42 per cent } 100 per cent

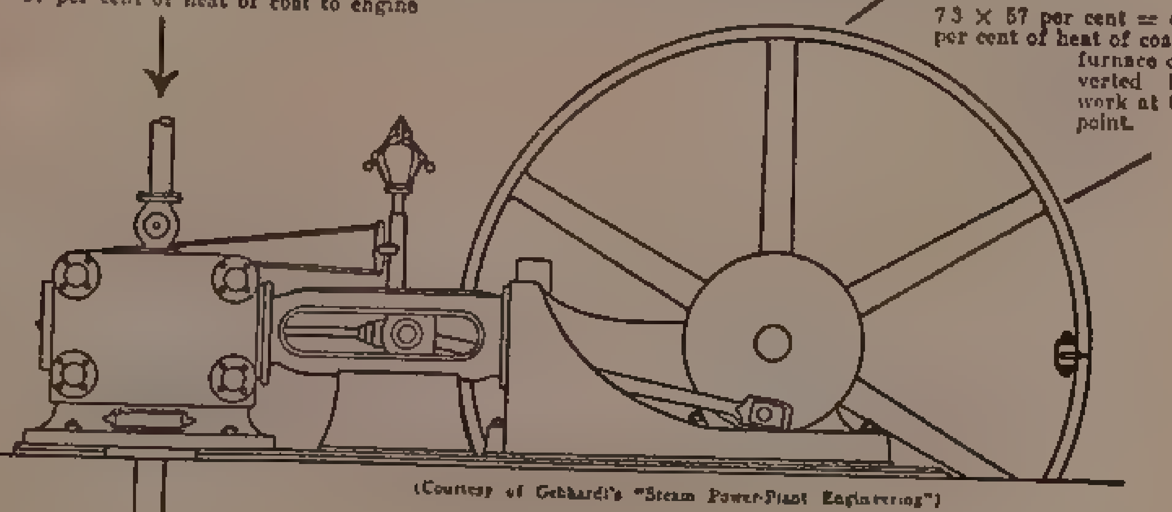
1 BOILER PLANT. RECEIVES 100 PER CENT OF HEAT OF COAL. DELIVERS 68 PER CENT OF IT TO STEAM PIPING



Efficiency of piping .... 98.3 per cent
Waste thru piping .... 1.7 per cent

98.3 per cent of 58 per cent = 57 per cent

2. STEAM PIPING. RECEIVES FROM BOILER 58 PER CENT OF HEAT OF COAL. DELIVERS 57 PER CENT TO ENGINE



Note:—The engine friction is about 5 per cent of its actual output, that is .05 × .073 = .00365, i.e. less than 4/10 of 1 per cent of heat entering engine, and is therefore eliminated from this analysis for the sake of simplicity.

This heat now enters the feed-water heater .... { Exhaust steam contains 92.7 per cent of heat in steam entering engine .... Or 92.7 × 57 per cent = 52.8 per cent of heat in coal to furnace .... This exhaust at 5.3 lb. back pressure contains 5 per cent of moisture .... } See note. See also foot note to Plate 3.

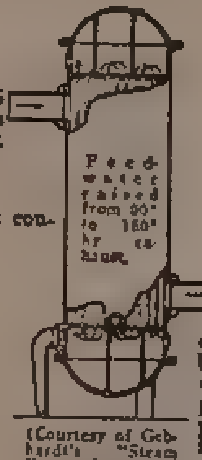
Engine efficiency calculation {
Heat in one lb. steam at 101.3 lb. Ga. press. (above 60°) .... 1,161 B.t.u.
Heat in 30 lb. steam at 101.3 lb. Ga. press. (above 60°) .... 34,830 B.t.u.
Heat in one H. P. hour without loss .... 2,546 B.t.u.

$$\text{Efficiency} = \frac{\text{out-put}}{\text{in-put}} = \frac{2{,}546}{34{,}830} = 7.3 \text{ per cent.}$$

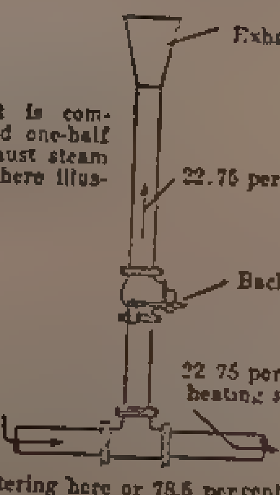
3. ENGINE PLANT. RECEIVES FROM PIPING 57 PER CENT OF HEAT OF COAL. DELIVERS 4.16 PER CENT OF THIS IN MECHANICAL ENERGY AND DISCHARGES 52.8 PER CENT IN THE EXHAUST*

52.8 per cent of heat of coal enters feed water heater, including 5 per cent moisture in exhaust steam—in "dry saturated" steam this amounts to 88 per cent of heat of steam entering engine or 86.5 per cent of heat of steam from boiler.

Heat absorbed by feed-water $\frac{90}{1161}$ = 7.76 per cent of heat contained in original boiler steam. This is returned to boiler. To allow for losses deduct 8 per cent from exhaust steam.

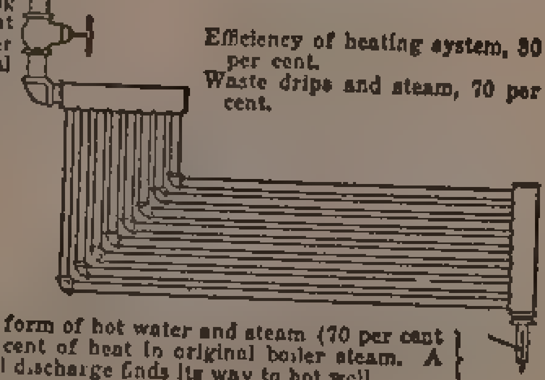


4. FEED WATER HEATER. RECEIVES 52.8 OF THE ORIGINAL HEAT OF COAL AND ABSORBS 7.3 PER CENT OF IT, LEAVING 45.5 PER CENT IN THE ISSUING STEAM



Note:—It is common to find one-half of the exhaust steam wasted as here illustrated.

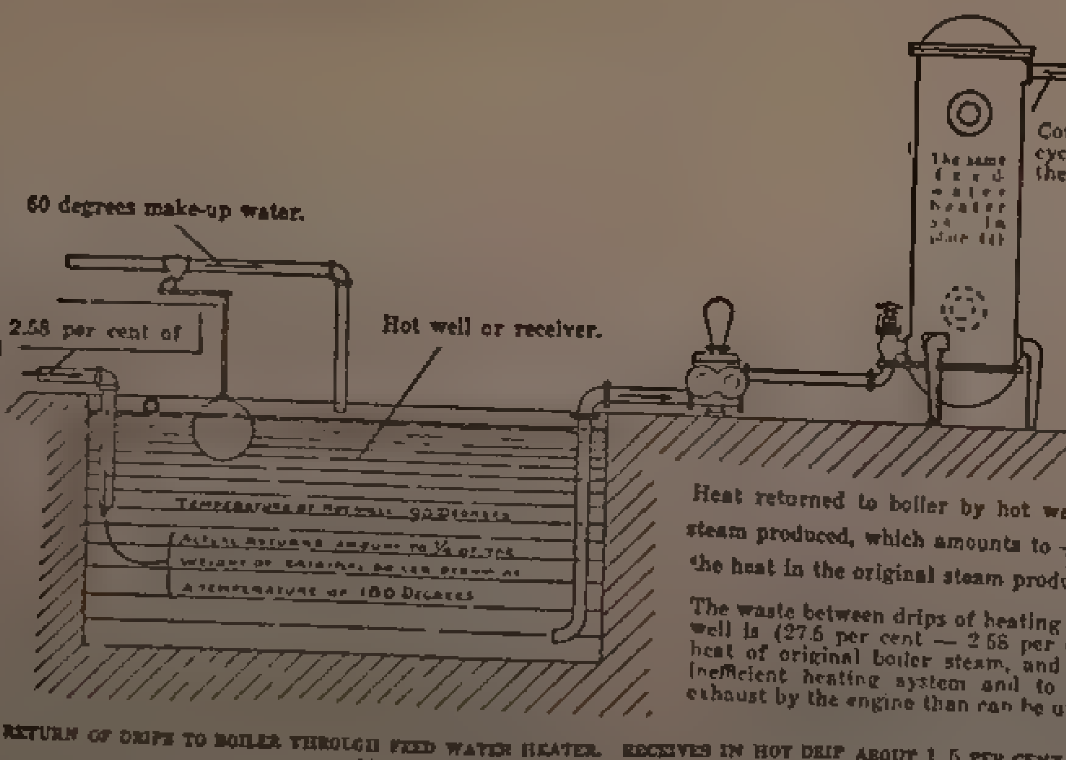
5. EXHAUST STEAM PIPING AND RELIEF VALVE. RECEIVES 45.6 PER CENT OF ORIGINAL HEAT OF COAL. DELIVERS 22.75 PER CENT OF IT TO HEATING SYSTEM AND WASTES 22.75 PER CENT TO ATMOSPHERE.



Efficiency of heating system, 30 per cent.
Waste drips and steam, 70 per cent.

Discharged by drip in form of hot water and steam (70 per cent of 39.25) = 27.5 per cent of heat in original boiler steam. A small part of this total discharge finds its way to hot well.

6. HEATING SYSTEM. RECEIVES 22.75 PER CENT OF ORIGINAL HEAT OF COAL. RADIATES 6.75 PER CENT OF IT AND DISCHARGES 16 PER CENT, OF WHICH A SMALL PART IS RECOVERED IN THE DRIP



Heat returned to boiler by hot well is 30 B.t.u. per lb. of steam produced, which amounts to $\frac{30}{1161}$ = 2.58 per cent of the heat in the original steam produced by the boiler.

The waste between drips of heating system and return by hot well is (27.5 per cent — 2.58 per cent) = 24.9 per cent of heat of original boiler steam, and is principally due to an inefficient heating system and to the production of more exhaust by the engine than can be utilized.

7. HOT WELL AND RETURN OF DRIPS TO BOILER THROUGH FEED WATER HEATER. RECEIVES IN HOT DRIP ABOUT 1.5 PER CENT OF THE ORIGINAL HEAT OF THE COAL AND RETURNS IT TO THE BOILER

Foot Note to Plate 3. *CALCULATION OF HEAT AND MOISTURE IN EXHAUST STEAM

Initial steam pressure .... 101.3 lb.
Heat in steam above 60 degrees .... 1161 B.t.u.
Absolute back pressure .... 5.3 lb.
Temperature of saturated steam at 5.3 lbs. .... 124 degrees
Heat in saturated steam at 5.3 lbs. above 60 degrees .... 1130 B.t.u.
Thermal efficiency of engine, assume .... 7.3 per cent
Disregarding radiation, one pound of exhaust mixture will contain 0.927×1161 .... 1076 B.t.u.
Temperature of water in this mixture at 5.3 lbs. back pressure 124 degrees .... 124 degrees
Heat above 60 degrees in each lb. of this water .... 164 B.t.u.
Let x = steam in one pound of exhaust mixture 1 — x = water in one pound of exhaust mixture
Then 1130 x + 164 (1 — x) = 1076 x = 0.944 = steam in exhaust mixture 1 — x = 0.056 = water in exhaust mixture

FIGS. 1. TO 7. CYCLE OF STEAM DISTRIBUTION AND SUCCESSIVE HEAT LOSSES IN A PLANT WHERE ALL STEAM GOES DIRECTLY TO THE ENGINES AND NO LIVE STEAM IS USED FOR HEATING.

are two common methods in factory power plants for the distribution of this energy: 1, direct mechanical transmission, by shafting and belting or rope drive; and 2, electrical transmission, which also involves usually a certain amount of mechanical transmission.

Now since there is a vast amount of confusion as to the relative advantages of these two methods and regarding the controlling factors, we shall examine into both systems with an illustration of each, together with their respective attendant losses and efficiencies. Before proceeding, however, let it be clearly stated that there is no *prima facie* reason for expecting great differences in efficiency from either system, but that the advantages of one over the other depend entirely upon local circumstances. As will be shown in our illustration, *for every loss in mechanical transmission there is a corresponding loss in electrical transmission,* and the particular conditions surrounding any given case will alone determine the advantages of the one over the other. Such advantages, it is true, may at times be very great indeed, but so much misinformation at present exists among the laity on this subject that a pair of analyses will be of service.

Referring now to our foregoing diagrams

of heat and steam distribution it will be remembered that only 7.3 per cent of the heat in the steam supplied to our engine was converted into mechanical energy or work. We have seen what in part became of the balance of the original supply of heat or energy from the coal pile, but at present our interest is confined to this 7.3 per cent of the heat of the steam which we now have in the form of work at our engine shaft. During our inspection of these examples of transmission it must be remembered that the variations in efficiency of either system under different conditions of operation and design are very great, and these cases are selected simply for the purpose of tracing their processes of efficiency and loss in a manner that is constant qualitatively but extremely variable when quantitatively considered.

For both cases we have at the engine shaft 7.3 per cent of the heat energy of the steam entering the engine, which represents an over-all efficiency of 4.16 per cent referred to the original heat in the coal. For clearness of analysis consider the energy at the engine shaft 100 per cent.

The analyses on pages 30, 31 give a fair representation of power distribution and loss on a percentage basis of the full or constant output of the main engine. Now while this

analysis holds true qualitatively at all times, yet there may be savings possible with an electrical system in cases where its use will enable the complete shutting down at times of departments of a factory for which otherwise it might be necessary to run long and heavy line shafts continuously, irrespective of actual power required.

On the other hand, if all machinery is to be run at full capacity continuously, in the case we have analyzed electrical transmission would save no steam at the engine nor any coal in the boiler room. This whole matter is capable of computation for any given set of conditions in advance of making contemplated changes; and in such calculation (which should be based upon actual test data) the cost factors must be carefully applied in order to learn the true commercial efficiency of the undertaking.

COMPARATIVE ANALYSIS OF MECHANICAL AND ELECTRICAL POWER DISTRIBUTION AND LOSSES

Mechanical Transmission

At engine shaft		100
Efficiency of main belt	0.97	
Loss at main belt	0.03	
Delivered at line shaft		97
Efficiency of line shaft	0.85	
Friction of line shaft	0.15	
Delivered at countershafts		82.5

Efficiency of other shafting and belting.	0.88	
Friction of other shafting and belting. .	0.12	
Delivered at machines.		72.6

Energy delivered at machines 72.6 per cent × 4.16 per cent = 3.02 per cent of the heat in the coal.

Electrical Transmission

At engine shaft.		100
Combined mechanical and electrical efficiency of dynamo.	0.90	
Loss at dynamo.	0.10	
Delivered on switchboard.		90
Efficiency of electrical mains.	0.98	
Resistance of mains.	0.02	
Delivered at motors.		88.2
Efficiency of motors.	0.88	
Mechanical and electrical loss in motors.	0.12	
Delivered at machines or short countershafts.		77.6
Efficiency of countershafts and belting on groups of machines.	0.92	
Friction on countershafts and belting on groups of machines.	0.08	
Delivered at machines.		71.4

Energy delivered at machines 71.4 per cent × 4.16 per cent = 2.97 per cent of the heat in the coal.

The principal teaching of our discussions thus far, however, is that we are able to analyze each successive step in the generation and utilization of energy in the factory power plant. The methods of performing this analysis will be discussed later, and it will be shown not only how we may actually determine the over-all and the individual ef-

ficiencies and losses in our processes, but the investigation of the preventable losses will be taken up in detail.

For the present let us realize, if we can, that no energy is really lost—it is only misdirected; that the principle of the conservation of energy as pointed out in our first chapter is a practical theory, and constitutes the very foundation of our present-day practice of power-plant investigation and improvement.

We have seen in the foregoing analyses that each successive stage in the chain of conversions and transmissions of energy has an individual efficiency of its own, and that the ultimate efficiency at the end of the chain is only the product of them all. Consequently it follows that a high ultimate efficiency can be obtained only from a thorough study of the individual efficiencies. The improvement that may be possible depends upon the percentage of preventable loss in each of these steps or stages. *The loss, or rather waste, in each step of conversion or transmission is simply the difference between 100 per cent and its determined efficiency.* The energy which constitutes this waste is all used in ways that can be determined, both as to location and extent. It is not lost except in so far as it affects the efficiency of

our plant. Since it is not lost, it can be traced and measured and analyzed by application of scientific knowledge. When thus investigated, that part of the waste which is preventable can be determined, and the cause being also definitely learned it can be eliminated, with the result that this preventable portion is saved and added directly to increase the old efficiency.

Now our general view of the operation of the typical factory power plant has brought out a number of facts which will guide us toward a logical method of investigation. Some of the more important of these features are here enumerated.

1. The correction of a preventable waste in the boiler plant resulting in a given percentage of fuel saved will, from a fuel standpoint, affect the *whole cost of power and heating* throughout the plant to the extent of this percentage. This fact is independent of any proportion of the boiler steam that may be used direct for heating or process purposes, and renders an investigation of the boiler room of primary importance. Furthermore, the waste in our average boiler plant is very great, 42 per cent of the heat in the coal, and it may be added that a large part of this waste is preventable. Thus it is possible and practicable to operate a factory

boiler plant at 70 to 75 per cent efficiency. In the latter case we should be able to add 17 per cent of the 42 per cent waste to the 58 per cent efficiency, and the result would be $17 \div 58 = 29.3$ per cent more steam for the same fuel, or for the same steaming capacity as before a saving in fuel of $17 \div 75 = 22.7$ per cent.

A plant owner who was retaining me to investigate his fuel problem, said "We can see just where our principal loss is. Those drips from our heating system are not returned to the boilers and that means a large waste." As a matter of fact the drips he referred to were from a heating system which condensed possibly one-third of his boiler steam. The drips at 180 degrees would contain 120 B.t.u. per pound or 40 B.t.u. per pound of steam at the boilers. Since a pound of his boiler steam contained 1,150 B.t.u., the return of these drips would represent a saving of only $40 \div 1{,}150$ or about $3\frac{1}{2}$ per cent of his coal bill.

He was making the common mistake of beginning his investigation at the tail end of the problem instead of starting in at the source of all his power and heating, the boiler plant. Upon centering the investigation at this point a preventable loss of 30 per cent

of the entire coal pile was located and corrected.

This incident is quoted merely to illustrate the principle involved in the logical method of attacking the factory fuel prob-

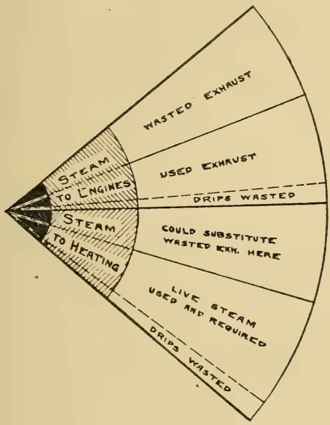


FIG. 8. DISTRIBUTION OF BOILER OUTPUT

lem. From the boiler plant as a centre and source, the uses and wastes of steam radiate outward like the vanes of an open fan, each blade or group of blades representing the proportion and consumption of steam in each department.

Figure 8 illustrates this, the black sector representing the entire output of the boilers.

Thus if we find a means of saving say 15 per cent of the "live steam," which in the illustration represents about 25 per cent of the total boiler steam, our saving will not be 15 per cent of the coal bill, but only 15 per cent of 25 per cent, or less than 4 per cent at the source of heat.

Thus are we enabled to see clearly why it is of first importance to subject the boiler plant to the most thorough and exhaustive investigation, and furthermore why this department should logically come first for our consideration.

2. The second fact of importance made clear by our general view of the problem as a whole is in connection with the very low efficiency of our steam engine. In the example quoted a simple Corliss engine has been selected as representative of general factory conditions, and 7.3 per cent thermal efficiency is a fair working average of the performance of this type. In this case our engine is run non-condensing and the question of the comparative economy of condensing naturally arises. We have seen that of the total heat that enters the throttle of our engine over 90 per cent escapes through the exhaust pipe. At what point will it pay to install a condenser, in order to improve the thermal efficiency of our engine as related

to the other expedient of better utilization of its exhaust heat? Why not put in a steam engine of the most economical type with the best kind of condensing equipment, and heat our factory with steam from a separate low-pressure boiler? What would be the result of this plan? Would it be best to throw out the steam engine entirely and substitute an oil engine, a producer-gas plant, or current purchased from a water-power or central-power plant?

What are the efficiencies of other prime movers compared to the steam engine? If they are better why not use them? A hundred and one questions like these, and more, arise from our look into the heat efficiency of the steam engine. And these questions thus inspired form the *raison d'être* of the great field of applied science comprising this department of our problem, second only in importance to the boiler plant.

3. We have noted, to the surprise of some of us no doubt, the tremendous amount of heat contained in the exhaust steam, although its pressure has been reduced from 101.3 pounds to 5.3 pounds and during this fall in pressure has developed the full horse power of our engine. We have seen that our plant is blowing away about half of this valuable by-product of the steam engine, and

of the other half only a fraction is utilized by the heating system into which it is fed. Here, indeed, are very great losses vitally affecting the consumption of fuel at the boiler plant.

We have seen that steam at the low pressure contains within 2 per cent as much heat as steam at 101-pounds pressure, disregarding the effect of the engine; and when the engine is included we still have some 90 per cent of the original heat of the steam in the form of exhaust.

This being true, why not cut out live steam entirely and substitute exhaust from the engine? A great saving may result from this measure if conditions permit. What then, are these limiting conditions? When by-product exhaust steam is worth practically as much as expensive live steam direct from the boilers, it is reasonable to assume that a complete knowledge of the characteristics of exhaust steam and of the factors connected with its utilization must constitute a most important element in our ultimate efficiency and consumption of fuel. *This in fact is so emphatically the case that the successful solution and minimum cost in many a factory power plant hinge directly upon the skillful handling of this feature of the problem.*

So closely connected with this matter that

it cannot be separated from it is the efficiency factor of the heating system for buildings and process work. Our typical analysis has shown a waste of 70 per cent in the *use* of steam in the heating system. We have also seen where this waste goes and how small a part of it is returned to the boilers. Therefore a careful study of preventable losses in heating systems is essential, and since these losses or wastes are measurable, as well as the utilized portion of the energy, we are able to grasp the problem with the simplicity of directness. This advantage is characteristic of our whole problem since it is completely governed throughout by the very beautiful and satisfying laws of the conservation of energy.

We have now a fair inkling of the underlying philosophy of power-plant investigation. I have stated that this philosophy has naturally evolved an efficient method for attacking any given problem, so that perhaps the next logical step in our discussion will be the statement of this method which is comprehensive for any given case.

Again, for the sake of clearness, a steam plant will be taken for an example, rather than risk possible confusion by an attempt to make statements both broad and definite enough to cover all cases. Let it be under-

stood, however, that the principles involved completely cover all imaginable types of power and heating plants.

Plan of Factory Power-Plant Investigation

1. Boiler Plant. Tests under actual working conditions will reveal the following information.

Combined Efficiency of boiler and furnace and a comparison of this efficiency with what they would develop under right conditions of operation and equipment.

Evaporation per Pound of coal as fired, of dry coal and of combustible.

Cost of Fuel for evaporating 1,000 pounds of steam.

Capacity of Boilers, i. e., horse power actually developed compared to their rated capacity or heating surface, showing whether boilers are over- or under-loaded for best economy. This will show whether the right number of boilers are being operated, whether too many or too few for lowest coal consumption.

Number of Heat Units in a Pound of Coal, ash, moisture, etc.

Total Waste of Heat of Coal. This is obtained by taking chemical analyses of the

gases resulting from its combustion, together with temperature of these escaping gases and of the air in the fireroom, and with analysis of the coal and other test data.

Heat Balance. A definite accounting of the expenditure of all the heat in the coal consumed. Thus the *Efficiency* is the percentage of this heat absorbed by the boiler and converted into steam. The balance of heat is then accounted for item by item to make up the full 100 per cent.

Load Fluctuation. As affecting type of boilers, kind of coal, and design and operation of furnaces.

Preventable Waste. An examination of the heat balance shows exactly what portion of the total waste may be reclaimed and added to the efficiency. Such portion may be collectively caused by excessive radiation, unburned or partially burned combustible, excess of air supply to the fires, leakage of cold air through setting, loss of carbon through grate or through cleaning operations, each one of which effects has a direct cause which can be eliminated through changes in either operation or equipment, and generally largely through the former. The special local causes producing these losses may be determined for elimination.

Cost of Evaporation with some other or cheaper kind of coal. Adaptability of present equipment for such change and its attendant cost of installation and equipment. This includes an investigation of all coals on the market, taking reliability of supply and fluctuation of prices into consideration, as well as any possible additional cost of firing or ash handling.

Coal and Ash Handling. An examination of this subject under local conditions with respect to the possibility of reducing cost of the work.

Finally, Recommendations based upon the complete finding of the boiler-plant investigation may be made in such manner that the maximum commercial saving which can be accurately predicted will result from their execution.

2. Steam Piping. The heat and fuel loss due to badly designed or uncovered steam piping are calculated accurately, with estimate of financial saving possible by improvement in these respects. Also economy due to safety of properly arranged steam piping, protection of engines from water, etc. Safety to operatives as well as plant depend upon safe boiler and engine piping and precaution against accident.

3. Engine Plant.

Test to Determine Efficiency of Engines, amount and cause of waste, including:—

Steam Consumption per Horse-power or Kilowatt Hour compared to good normal consumption for the particular type of equipment. If below this efficiency the amount of preventable loss is known, and its location is determined by indicator and leakage tests, as well as friction and back-pressure tests on the engine and efficiency test on the electric generator if one is driven by the engine.

Thus the amount, location and cause of the preventable losses are discovered and correct remedy can then be definitely recommended.

Any saving that might be possible by substituting a more efficient type of engine will now be calculable, the possible use of exhaust steam to figure in this consideration.

Fuel Cost per Horse-Power or Kilowatt Hour can now be obtained (the boiler test having been made) and should be decreased by a true allowance for any part of the exhaust steam from the engine which is taking the place of an equivalent amount of *live* steam that would otherwise be required.

Value of Condensing versus Non-Condensing is definitely estimated with consideration of all surrounding and local conditions.

Boiler Pressure as affecting engine economy accurately calculable.

Superheating steam at boilers or with separately fired superheater investigated for local conditions and thermal and commercial efficiencies decided with proper consideration of design and construction of engines and piping and percentage of boiler steam to engines or carried long distances. Effect upon dryness of exhaust steam with bearing on heating.

Determination of relative values of high-, low-, or mixed-pressure turbines, compound or simple engines, etc., for the local requirements.

Day and Night and Summer and Winter Loads as affecting number and type of engines.

Recommendations. These are made only after investigation of both boiler plant and heating system.

4. Heating System, including steam for process work.

Determination of that Part of Boiler Steam at present used direct.

a—For heating, day and night, summer and winter.

b—For process, day and night, summer and winter.

Determination of that Part of Boiler Steam which goes to engine and pumps.

Determination of that Portion of Exhaust Steam which is efficiently utilized and that part which is wasted.

Amount of Direct Steam that could be replaced by exhaust now available, or which could be made available and commercial economy of such change worked out.

Efficiency of Present Heating System as found, and amount of preventable waste referred to coal consumption.

Crediting to engine performance that portion of their heat given to the heating system, as affecting cost of fuel for power only.

Recommendations on Heating based not only on separate investigation of same but also as influenced by the result of examination of Boiler and Engine questions as per previous headings.

5. Cost of Power. This is of especial importance when a proposition for buying electric current from central or water-power station demands consideration. The full data obtained on boiler plant, engines and heating enable an accurate determination of power cost per kilowatt or horse-power hour by combining such data with full operating costs which include labor, repairs, interest, depreciation and insurance. The utiliza-

tion of exhaust steam is a large factor in this result.

Present Cost of Power determined as above.

Reduction in Existing Cost of Power. The amount by which this present cost per kilowatt or horse-power hour can be reduced depends entirely upon the information regarding preventable waste throughout the whole plant as determined by the foregoing investigation.

The cost of making the plant efficient should be added as an interest charge, and then this new cost of power that can be produced in the improved factory power plant is directly comparable to the cost of current as offered by the electric company, providing certain charges connected with the price per kilowatt-hour of the latter are added to it for a just comparison.

Thus it is practicable to determine precisely for any given factory power plant at what price it will pay to contract for the purchase of outside electric power.

We have seen that the science of power-plant improvement depends upon the principles of the conservation of energy, that the logical application of these laws has evolved a normal and comprehensive method for the determination and measurement of prevent-

able fuel waste; and in accordance with the method thus developed we shall now consecutively consider the individual efficiencies constituting the complete cycle which we have graphically illustrated, whose combined product constitutes the ultimate efficiency of our factory power plant. In this proceeding our object will be not only to describe the theory of testing each of these specified efficiencies, but also to give examples of such tests taken from actual reports of this nature and to show in a practical manner some of the resultant losses that are thus determined together with the means for their correction. In this treatment the boiler plant will logically receive our first attention, and since instances from actual investigations will be quoted commercial efficiency will be seen to be the ultimate object of the work throughout. But it will be equally apparent that the necessary and only available means for a complete knowledge of this financial value is the determination of thermal efficiency in accordance with the laws of the conservation of energy.

Chapter III

THE BOILER PLANT

INVESTIGATIONS for determination of waste in the boiler plant must not only be made in such a manner that the efficiency and losses of operation will become known, but they must be performed with such thoroughness that the exact location, cause, and extent of the preventable portion of all losses will be made evident.

Furthermore, the preventable losses must be determined with equal thoroughness from the standpoint of commercial efficiency. All these desirable results can be obtained. Furthermore, they *are* obtained in actual practice, and without disturbance to the regular operation of the plant. They are naturally followed by improved efficiency and a reduced cost for steam, since the existing faults are definitely determined and measured, eliminating all guess work in the recommendation of proper means for their correc-

tion. It requires no stretching of the truth to say that boiler-plant improvement by this method of investigation becomes an exact science, and one of great practical value.

After getting acquainted first with the executive and operating members of the boiler-plant department, the investigator makes arrangements for conducting boiler tests which shall give the results of an average performance of the boilers and furnaces under average working conditions. The chief engineer is requested to urge upon his firemen that they shall pay no attention whatever to the boiler testing but shall perform their duties in their accustomed everyday fashion. This is very important; for if, as sometimes happens, the firemen receive the impression that they are in some way being spied upon, they will try to make a record during the test, in which case the results will usually be above the average of daily practice. If through neglect of the proper precautions this should happen, it alters the test results and the determination of preventable waste. But even in this circumstance, the difference in the calculations of the expert is on the safe side. That is to say, if the plant is operated at higher than usual efficiency, the possible savings predicted by the investigator will be proportionately smaller, so that

when his recommendations are put into effect the actual savings thereby produced will be larger than his original estimate. This, however, is not desirable and the necessary precaution should be observed for the prevention of its occurrence.

As a further preliminary measure, the object and method of the whole investigation should be fully explained and discussed with the operating engineer. When he is the kind of man who is truly interested in the efficiency of his company's plant and is ambitious for his own mental and pecuniary advancement, he will be anxious to aid in the work and to gain all the knowledge and information from the tests that he possibly can. If ocasionally this is not the case, then appropriate measures are required according to the circumstances.

Since combustion consists of a chemical reaction of gases and elements with the oxygen of the air, the efficiency of its operation is not visible to the eye. The steam gas in the boilers which is the object of the combustion is also invisible. Consequently the average factory-plant management is quite at sea regarding the efficiency of its boilers and furnaces, even though it may be very modern in its efficiency methods when dealing with the processes and products of the

manufacturing departments. The efficiency of a boiler plant is a hidden matter. It lies beneath the surface. Nothing less than a scientific test will reveal the usually great losses that are taking place. These losses, which are largely preventable, are quite independent in any degree of the external appearance of the equipment itself, and such matters as white-tile finish and polished brass are frequently and justly comparable to the "goodly apple rotten at the core". In one manufacturing plant which I investigated the boiler house was its particular pride and joy, and no suspicion had rested upon it as the possible source of a great waste which baffled the owners. The plant was indeed pleasing to the eye in every particular, but when subjected to test exhibited a very different aspect from the pocket-book point of view. The boilers were found to be developing only half of their rated capacity, and although the total requirements were only in the neighborhood of 600 horse power the commercial-efficiency test discovered a preventable loss of $7,000 per year in fuel expense.

In addition to the necessity of boiler testing, this example suggests the value of placing a plant under a strict efficiency system which shall daily reveal to the management

the economy of its operation. This feature will be carefully treated in its proper place.

Very frequently unreliable reports are made as to the performance of boilers, and large indeed is the number of false or misleading statements of this kind. The average layman accepts as criterion a statement or determination of "water evaporated per pound of coal". He makes no distinction between *actual evaporation* and *equivalent evaporation from and at 212 degrees.* These different ways of reporting a test may affect the resulting figure as much as 20 per cent, and if the business man is not sure of how his test was made and just what the figure means, he would do well to avoid what may prove to be an expensive mistake by taking the wise precaution of finding out.

Still another common error, and as serious a one, consists in taking the figures which represent evaporation per pound of *combustible* for the evaporation per pound of *coal.* This leads to much trouble because coal and combustible are not the same thing. The combustible portion may be only 80 per cent of the weight of the coal. Consequently the evaporation based on combustible is always a correspondingly higher figure than that based on the coal. Furthermore, there are two ways of reporting the amount of com-

bustible, the chemical and the mechanical, each with its own result. Consequently when we consider "*actual*" evaporation and "*equivalent*" evaporation as each being commonly referred *both* to *coal* and to *combustible,* we have *four* correct though incomplete methods of stating boiler results. Add to this the fact that the amount of combustible is obtained in the two different ways just mentioned, and we have six different methods of reporting evaporation. There are still two more ways, viz: actual and equivalent evaporation individually based on the coal *as fired,* which contains moisture sometimes as much as 10 per cent, altering the results and their significance to a marked degree.

Enough has been said on this point to indicate the trouble which managers are likely to have in understanding and interpreting correctly the value and meaning of the term "evaporation". And yet it remains true that when in some manner they have obtained a single one of the numerous evaporation figures from their boiler plant, they frequently imagine that they know the degree of economy with which their boilers are working. But as a matter of fact they have no such knowledge, even if the determination is entirely correct, which is seldom the case.

For the economy indicated by an evaporative result depends entirely upon the heat value of the fuel. Thus an "equivalent" evaporation of 8 pounds based on dry coal will represent the fairly high boiler and furnace efficiency of 70.5 per cent if a pound of the coal contains 11,000 heat units; but the same evaporation with a coal of 14,500 B.t.u. will exhibit the poor efficiency of only 53.5 per cent, a vast difference indeed. And so the evaporative result of a boiler means nothing whatever as indicative of its efficiency unless the heat value of the fuel be taken into our calculation. Now, by referring to previous chapters it will be remembered that efficiency is always the useful output of energy (or heat) compared to the input, i. e., Output ÷ Input = Efficiency. In the case of our boiler and furnace the heat in a pound of coal represents the input and the heat in the steam generated by each pound of coal is the output. Now the heat put into a pound of steam varies with the temperature of the feed water and the pressure of the steam in the boiler. For convenience and uniformity of expression, mechanical engineers have adopted a unit of evaporation which is simply the amount of heat that is necessary to evaporate a pound of water from a feed-water temperature of 212 degrees into steam

at the same temperature and the corresponding or atmospheric pressure, and this actual amount is 970.4 heat units (B.t.u.). Consequently, by a simple calculation involving the factor of evaporation, all actual evaporations under varying temperatures and pressures are reduced to their equivalent evaporation from and at 212 degrees, and each pound of equivalent evaporation represents 970.4 B.t.u.

Thus the efficiency of a boiler and furnace is determined by dividing the "output" or pounds of equivalent evaporation, multiplied by 970.4 B.t.u., by the B.t.u. in a pound of coal producing that evaporation.[1] With correct data in hand it is therefore a simple matter to obtain the efficiency of our boiler plant, and when thus obtained it forms a true and *the only true measure of economy.*

Irrespective of the value of the coal, the pressure of the steam, the temperature of the feed water, the kind of apparatus, and of any and all conditions whatsoever, the thermal efficiency of the boiler and furnace

[1] The factor of evaporation is found for a given combination of steam pressure and feed-water temperature by dividing the actual heat in a pound of this steam above its feed temperature by 970.4. Thus the formula becomes:

$$F = (H - h) \div 970.4 \text{ in which}$$

H = total heat in the steam above 32 degrees at the operating pressure; h = total heat in the feed water above 32 degrees; 970.4 = latent heat of evaporation at 212 degrees; F = the factor of evaporation.

forms the one and only criterion of the degree of economy or waste.

Having obtained the efficiency of the plant, we shall immediately know whether its performance is above or below good practice. If it happens to be below (as is generally the case) it becomes necessary to study and to analyze the *inefficiency* and the *waste.* These are always the differences between the carefully determined percentage of efficiency and 100 per cent. Thus we know exactly what part of the heat of the coal is converted into useful steam and what portion is lost or wasted.

If we had to stop at this point in our determinations, our knowledge and consequently our ability to improve would be limited. But owing once more to our natural laws of conservation, we are enabled to measure all of the losses as well as the percentage of heat utilized. This analysis of waste, together with the efficiency, must invariably total up to the original heat in the coal at the beginning of the operation. Such a balance of efficiency and loss is known as the "heat balance." With the heat balance before us, we shall not only know where the greatest losses occur but we shall also know what causes them, and consequently how to reduce them. We are therefore able to work intelligently

and can predict with great certainty the amount of saving that will be produced by making the changes of operation or design that are correctly indicated by the character and extent of the established losses.

The actual determination of the heat balance involves in the first place a thorough and complete boiler test, together with a knowledge of chemistry, for combustion involves the conversion of chemical energy into heat energy.

HEAT BALANCE OF A BOILER AND ITS FURNACE

Total Heat Value of one pound of combustible—100 per cent

	Per cent
Heat absorbed by the boiler—useful	x
Losses accounted for:	
Loss due to moisture in the coal	a
Loss due to moisture formed by burning of hydrogen	b
Loss due to heat carried away by dry chimney gases	c
Loss due to incomplete combustion of carbon	d
Loss due to incomplete combustion of hydrocarbons	e
Loss due to incomplete combustion of sulphur	f
Loss due to moisture in the air and to radiation	g
Loss due to combustible dropping through grate and removed in cleaning fire, also heat so removed	h
Total of losses plus useful heat	100

It is aside from the purpose of this book to enter upon a detailed discussion of boiler testing, of combustion or of steam, for any one of which great volumes would be required. It is simply desired to indicate those sources of knowledge upon which it is necessary to draw freely for the production of a complete and informative boiler test. Having done so in a brief manner let us examine the items which make up our heat balance, which has been the practical object of our test.

In order to obtain the data from which to calculate our heat balance it is necessary to make an average day's run on the boiler or boilers and to take careful and accurate records of water, coal (including its weight, moisture, calorific value and chemical analysis), readings of all essential temperatures, pressures, and draft tests. With a proper testing equipment this is not so complicated and difficult as it may appear and, when it becomes understood that by such means alone can the preventable losses be determined, the application of this particular branch of practical science will increase with even greater rapidity than has marked its progress in recent years. If we carry this out thoroughly, then when we have made our efficiency test, which may indicate a poor economy, the heat

balance tells us exactly *why* our result was so poor.

If then we add to our heat balance the factor of commercial economy, involving a determination of the comparative values of all of the available fuels on the local market, we shall have covered the ground so completely that we shall have definite knowledge of the correct solution of our problem. We shall know how many dollars per year can be saved, and how to go about producing that saving without guesswork as to any of the factors involved.

In order to demonstrate the actual working of the heat balance I will quote from a section of one of my reports on a small factory boiler plant. The figures and notes follow:

HEAT BALANCE BASED ON COMBUSTIBLE, A. S. M. E. CODE

Combustible = 15,445 B.t.u. per lb.

	B.t.u.	*Per cent*
1—Heat absorbed by boiler......	8,155	52.80
2—Loss due to moisture in the coal	22	0.14
3—Loss due to moisture formed by burning to hydrogen (5 per cent H assumed in combustible).................	544	3.52

	B.t.u.	*Per cent*
Brought forward	8,721	56.46
4—Loss due to heat carried away in dry chimney gases (weight of gas per lb. combustible × 0.24 × [T — t])	3,700	24
5—Loss due to incomplete combustion of carbon (C to CO).		0
6—Loss due to unconsumed hydrogen and hydrocarbons, to heating moisture in air, to radiation, carbon in discharged ash and unaccounted for (by diff.)	3,024	19.54
TOTALS	15,445	100.00

Notes on Heat Balance

The loss due to heat carried away in dry chimney gases is shown in Item 4 in the above balance. This is computed according to principles laid down in Chapter XII on Combustion. From formulas in this Chapter it is determined that by improving the firing so as to increase the CO_2 from 5.35 per cent to 10 per cent, the dry chimney loss will be reduced from 24 per cent to 11.7 per cent of the heat of the combustible. That is to say, 12.3 per cent of the heat involved will be added to the old efficiency of 52.8, making the new efficiency 65.1 per cent. The saving in coal for the same amount of steam production will be $\frac{12.3}{65.1} = 19$ per cent while the gain in steam for the same coal consumption as at present will be $\frac{12.3}{52.8} = 23.3$ per cent.

The method of the above determinations was as follows. From the composition of the fuel, the chemical air requirements were obtained with the formula

$A = 11.6 + (34.8 \times H_a)$. The corresponding or maximum CO_2 was obtained from the formula $P = \frac{152 \times 21}{152 + 361\ H_a}$. The weight of the dry products of combustion was found for 5.35 CO_2 and $10 - CO_2$ by means of the formula $W_d = 11.6 + 1 + A_e + 0.77 (H_a \times 34.8)$ in which $H_a = H - \frac{O}{8}$ and for which A_e is obtained by solving for it in the formula $P = \frac{11.6 \times 21}{11.6 + 26.7\ H_a + A_e}$.

With W_d known for the two conditions of analysis, the dry gas chimney loss is computed for each case and the reduction of this loss by the increase of CO_2 is directly added to the old efficiency.

Simply improving the combustion would create this gain, but there would also be an attendant gain from reduction of radiation losses by firing a single boiler instead of two boilers. Consequently the above saving is a safe figure, especially as no fancy efficiency is called for.

As a check upon these figures a 23.2 per cent gain in evaporation over the present evaporation of 7.3 would give 9 pounds of water per pound of coal from and at 212 degrees.

This evaporation of 9 pounds would represent a combined efficiency of $9 \times 970.4 \div 13{,}400 = 65$ per cent efficiency.

To give this result in terms of actual evaporation under observed conditions with the city water temperature as tested, the above result would correspond to $9 \div 1.18 = 7.62$ pounds instead of the evaporation found in test amounting to 6.09 pounds.

The change in the method of firing as specified was put into immediate effect, and the results obtained exceeded the safe prediction that the heat balance made possible. The

actual saving was over 25 per cent of the coal formerly consumed.

So much for thermal efficiency pure and simple. The commercial efficiency of the case was then calculated and I again quote from my report in this regard.

Anthracite screenings in this particular locality, however, seem to offer a much more attractive solution of the problem.

Good screenings or D & H birdseye contain 11,000 to 12,000 B.t.u. per pound, as compared to 13,400 B.t.u. in your present $3.50 soft coal.

The delivered prices on the former according to your information run from $0.90 to $1.40 per ton. The sample given me of the $1.40 coal from the X. Y. Coal Co. appears to be a very high grade of this fuel which, however, will vary more or less.

Now in order to make the following calculation a safe one I will figure on $1.40 instead of $0.90 per short ton, and on only 11,000 B.t.u. per pound.

Thus we have the following comparison with your present coal, moisture content assumed to be the same in each coal.

Table of Coal Value Comparisons

Present coal, bituminous $3.50 per 2,000 pounds delivered, 13,400 B.t.u. per dry pound.

Anthracite screenings, $1.40 per 2,000 pounds delivered, 11,000 B.t.u. per dry pound.

Cost of 1,000,000 Heat Units (B.t.u.)

Present coal, bituminous

$$\frac{\$3.50}{2{,}000 \times 13{,}400 \text{ B.t.u.}} = \$0.1305$$

Anthracite screenings

$$\frac{\$1.40}{2{,}000 \times 11{,}000 \text{ B.t.u.}} = \$0.0637$$

Based on a safe heat value and the highest price

per ton, the same amount of heat would cost you \$0.064 with the screenings as compared to \$0.1305 with the present soft coal.

Consequently if you burned the anthracite screenings at the same low efficiency now found to be obtaining with the soft coal, your coal bill would be $0.0637 \div 0.1305 = 48.8$ per cent of present bill, or a saving of 51.2 per cent per year of the present expense.

This figure, however, 51 per cent, does not represent the full saving that you can obtain by a substitution of this fuel, for the reason that instead of burning same at your present efficiency you would be able easily to burn it at an efficiency of 65 per cent with a properly designed equipment. I have designed two plants for this fuel which are operating at an even higher efficiency than this.

Your cost of evaporation with 11,000 B.t.u. screenings at \$1.40 per 2,000 pounds delivered, at 65 per cent efficiency, would be (at the rate of 7.37 pounds evaporation from and at 212 degrees)

$$\frac{1{,}000}{7.37} \times \frac{\$1.40}{2{,}000} = \$0.095$$

per 1,000 pounds of steam from and at 212 degrees.

As a further comparison and check upon these statements, one of my plants has a cost of evaporation of \$0.10.

Your present cost of evaporation is \$0.244. Consequently by taking the proper steps you have a coal cost saving waiting for you amounting to $(0.244 - 0.10) \div 0.244 = 59$ per cent of your present yearly coal bill. That is to say, about 60 per cent of the coal bill can be saved on a conservative estimate.

It will be remembered that this estimate is figured on 11,000 B.t.u. instead of 12,000 B.t.u. coal, and on a price for same of \$1.40 instead of a possible \$0.90.

Since the original writing of this section

all these changes have been carried out and have now been in operation about a year.

The client now reports his average cost of evaporation at about 9 cents instead of his original fuel cost of 24.4 cents. That is, his *saving* by these changes amounts to *63 per cent* of his former coal bill.

Thus we have an example from an actual case of the application to the boiler plant of the method of investigation logically evolved by, and dependent upon, the laws of the conservation of energy. We have introduced the money factor in its relation to thermal value, with the result of discovering a saving of over 60 per cent of the original coal bill.

It is the duty of the investigating engineer to translate into physical and financial terms the items of the heat balance, the determination of which forms the first part of his task. These items of wasted heat are directly traceable to certain causes which vary in number and kind as the case may develop, but in order to obtain a working idea of how this translation is effected let us take our heat balance and run over a brief description both of the determination and of some of the practical means of eliminating or reducing these losses.

We have already discussed, with sufficient

amplification for our purposes, the method of calculating the efficiency or useful percentage of the heat in the coal as determined from part of the data observed in the boiler test. We shall now consider each item of loss in turn, using the simplest form of heat balance based on combustible, and for this purpose (which is simply for illustration) we shall follow the form of the heat balance last quoted.

The heat value of the coal in this case was 13,400 B.t.u. per pound, dry. The ash in this coal by analysis was 13.24 per cent. Consequently the combustible matter was only 86.76 per cent and its heat value per pound must therefore be 13,400 B.t.u. ÷ 0.8676 or 15.445 B.t.u.

The first heat loss is that due to moisture in the coal and is numbered "2" in the balance. This water has to be raised from the temperature of the air in the fireroom to the temperature of the hot gases leaving the boilers, and in so doing is converted into superheated steam, thus involving the large item of its latent heat, i. e., 970.4 B.t.u. per pound. In addition, sufficient heat must be added to each pound of water to raise its temperature from that of the fireroom air up to 212 degrees, and again from this point in the form of steam gas up to the chimney tempera-

ture. All of these factors are taken as part of the boiler-test observations, so the loss is readily calculated. In this case it amounted to only 22 heat units for each pound of combustible or 0.14 per cent of the heat to the furnace.

The second loss is due to moisture formed by the burning of hydrogen to water or steam gas, and the heat carried away by each pound of such steam is calculated in the same way as just described for the moisture in the coal. It is first necessary, of course, to determine how much of this steam is formed per pound of combustible fed to the furnace. This is obtained by taking the percentage of hydrogen in the fuel and performing a simple chemical calculation. In this case our hydrogen moisture loss amounted to 3.52 per cent of the heat of the combustible.

The next or third loss is the amount of heat that is carried away up the chimney by the dry hot gases of combustion. This is commonly called the chimney loss, and is of the utmost importance in boiler practice. Its calculation depends upon the temperature and the weight of these gases as they leave the boiler. Here again we must resort to chemistry, and during our test we have taken a large number of samples of the flue gases and have subjected them to analysis in a chemi-

cal apparatus which determines the volumetric proportions of CO_2 (carbonic acid gas, the product of complete combustion of carbon), O (free oxygen that has remained uncombined or has played no part in the combustion), CO (carbon monoxide, or half-burned carbon) and nitrogen, the inert portion of the atmosphere which acts merely as an unavoidable diluent and cooling agent of our combustion gases. From these data we are able to determine the amount or weight of air that has been supplied per pound of combustible, and consequently the weight of the resulting products of combustion. This chimney-loss computation includes leakage of air through the boiler settings. Chemistry and physics give us the specific heat of these various gases, and knowing their temperature and their weight, these three factors are multiplied together and the resulting product is the number of heat units in the dry chimney gases for each pound of combustible supplied to the furnace.

In this particular case the chimney loss amounted to 3,700 B.t.u., or 24 per cent of the heat of the combustible. The chemical analyses made during the test also showed that about 43 pounds of air were being supplied to the furnace for each pound of combustible, whereas in good practice only 18

pounds are required. This great amount of surplus air beyond the combustion requirements was absorbing heat from the coal and carrying it away up the chimney, instead of allowing it to be absorbed by the boiler. Hence it was plainly indicated that this air supply should be reduced. Now other data from our boiler tests showed a very low rate of consumption of coal to the square foot of grate surface. That is, the grate was too large for the amount of coal to be burned, and consequently admitted too much air to the fire. Here, then, was the cause of the large chimney loss, with plain indications of what the cure should be. This cure was put into effect by reducing the grate surface more than 50 per cent and improving the firing, with the result that the air supply was reduced to less than one-half and the weight of the dry chimney gases was reduced from 43.8 pounds per pound of combustible to 21.5 pounds. The new chimney loss would then become $21.5 \div 43.8$ of 24 per cent or 11.7 per cent, representing a gain of 12.3 per cent to be added to the old efficiency of 52.8, making the new efficiency 65.1 per cent. The saving in coal would then be $(65.1 - 52.8) \div 65.1 =$ about 19 per cent. As a matter of fact, the effect was greater than this prediction which was on the "safe side" and the true saving

was over 25 per cent of the coal originally consumed for the same output of steam.

This case of too great an air supply is a very common one. Most people think that good combustion is simply a matter of ample air supply. As a matter of fact, more waste of coal is likely to occur from an over supply than from too little air.

In this particular case I have quoted more smoke was produced after the combustion was improved than before. This demonstrates what is so poorly understood, that smoke may indicate less waste than no smoke. A smokeless chimney may be produced by flooding the fire with air, but this entails a heavy increase in the chimney loss with a consequent waste of coal. This does not mean that it pays to make smoke, although for a given furnace and set of conditions it may be so. In other words, there are cases when the losses due to incomplete combustion as indicated by smoke may be less than the losses due to the admission of sufficient air to ''kill'' the smoke. This is the weak point of many so-called ''smoke-consumers''. The science of the matter lies in the production of perfect combustion, which is necessarily smokeless and which at the same time involves a correct proportioning (neither an excess nor a deficiency) of air to the fuel.

The production of such combustion is not entirely controlled by the simple matter of air admission, but to as great an extent by the intelligent design of the furnace itself. This involves the maintenance of high temperature and the thorough mixing of the air with the combustion gases and with the carbon, and constitutes a science in itself.

The fourth loss in our heat balance (item No. 5) is occasionally by the incomplete combustion of the carbon of the coal. When carbon burns completely it forms CO_2, which is carbonic-acid gas or carbon-dioxide, and the heat generated by this union with oxygen is 14,600 B.t.u. for each pound of carbon so burned. When carbon burns incompletely it forms CO, or carbon monoxide, which upon addition of more oxygen burns to CO_2. But since the burning of carbon to CO develops only 4,450 B.t.u. for each pound of carbon so burned we have lost $(14{,}600 - 4{,}450) = 10{,}150$ B.t.u.

Now from the analyses of the chimney gases made during our boiler test, we can compute how much of the carbon of our combustible has been incompletely burned to CO and consequently how much heat has been lost by this cause. In this special case with the great excess of air present the loss from this source was *nil*. In some boiler tests,

however, a large loss is indicated. It may be due to insufficient air supply, and if this is the case it will be so indicated by the very small excess of free oxygen as shown by the flue-gas analyses. If sufficient oxygen is present, then the loss is chargeable either to lack of thorough mixing of the air with the fuel and combustion gases, or to low temperature in the fire. These effects are traceable directly to furnace design and the method of firing, and when thus investigated can be corrected to the direct improvement of our efficiency as would be indicated by this item in the heat balance.

The next loss to be considered forms a part of item No. 6 and is that due to the direct removal of combustible matter. This is caused first by unburned fuel dropping through the grate into the ashpit. The loss thus occasioned is determined by analyzing the ash for its carbon content. Its weight multiplied by its heat value equals the loss. Another item of this waste occurs when incandescent clinker is removed from the grate during cleaning operations. In addition to the unconsumed carbon in this hot substance, a certain amount of heat is contained in the heated mass, which can be computed by means of its temperature, weight and specific heat.

In further sub-division of our last item there is the loss due to heating the moisture in the air. This is so small that in commercial work it may be neglected. The waste due to unburned hydrogen and hydrocarbons from the coal may be a large percentage, depending principally upon the volatile constituent of the fuel and how it is handled. This loss together with that due to radiation is generally found by deducting all the other items from 100 per cent, this being the simplest and most direct solution.

As hard, or anthracite, coals contain very small percentages of volatile matter, the loss due to escaping hydrocarbons is naturally small. But certain soft coals contain as much as 40 per cent of volatile material consisting largely of marsh gas (CH_4) which has a high heat value. When a shovelful of such soft coal is thrown into a hot furnace the greater part of this percentage of volatile matter at once becomes freed from the coal, and the generation of this valuable fuel gas is so rapid that unless the furnace conditions are right for its combustion most of it passes out of the boiler and up the chimney as a dead loss. This action is generally accompanied by smoke, which consists of unconsumed particles of visible carbon and tarry matter escaping with the invisible but valu-

able fuel gases. The smoke itself is an insignificant loss compared with the gases which frequently accompany it. The study of this loss teaches us to fire our coal a little at a time, so that there will be sufficient heat and temperature in the surrounding parts of the furnace to cause its ignition and enough air to permit its complete combustion. This results directly in reducing the smoke and the hydrocarbon loss at the same time, which brings us again to the matter of furnace design and operation. Waste of fuel may be caused by unintelligent handling of fires in a very good furnace, while good firing will not give best results in a bad furnace. Thus with the much contracted combustion space found in many settings it will be very difficult for the best fireman to get high efficiency, especially under forced conditions, while a scientifically designed furnace is no guarantee of good combustion unless intelligently handled.

While we shall not be allowed to undertake a discussion of the absorbing subject of furnace design, it will at this time be *apropos* to state a condition which vitally affects the loss due to unburned hydrocarbons from soft coals, for it is desired to learn in some measure what are some of the common methods of reducing the losses which have been

located and measured in our heat balance.

Aside from the loss due to excessive air supply as evidenced in the "chimney loss" item, the greatest waste in the burning of soft coal appears in the escape of a large percentage of its volatile constituent immediately following the introduction of the coal to the furnace or the slicing of the fire. Either operation liberates large volumes of the hydrocarbon gases, which ordinarily flow out of the furnace with great rapidity under the influence of the draft. The time element is of the utmost importance, for if these gases could be held in the furnace a sufficient length of time for thorough heating and diffusion with the oxygen, the greater portion of them would undergo combustion with a consequent development of the heat otherwise wasted. One of the very simple and effective means employed to accomplish this result is to provide large draft areas over the fire and in the combustion chamber. With a given flow of gases and air their velocity will be inversely proportional to the cross-section of the passage. Thus these volatile constituents can be retained in hot combustion spaces for double the time they formerly spent. Lowering the grate to bring the fire farther away from the boiler and deepening the combustion chamber often produces the desired re-

sult, which may be further benefited by the intelligent design of proper brick arches or baffle walls to aid the mixing action before referred to.

A great deal of misunderstanding exists in regard to the effect of placing the grate further away from the boiler. It is common belief that the nearer the fire is to the boiler the greater will be the resulting efficiency. But the very opposite is true within reasonable limits, especially with soft coal, and even with anthracite it is advantageous to have the grate much lower than is found in general use. The trouble is that people do not distinguish between the diametrically opposed functions of the boiler and furnace. The duty of the furnace is to produce complete combustion of the fuel, and for this purpose it must develop a maximum of heat at a maximum temperature. The boiler on the other hand is essentially a heat *absorber*, and to be efficient must reduce the temperature and the heat developed by the furnace to a minimum. Consequently the boiler if placed directly in the flames of the furnace reduces the temperature and retards the combustion by cooling the gases below their ignition point, which allows them to escape unburned. It follows that the two functions of furnace and boiler must be sufficiently

separated to allow complete combustion and full development of the heat from the fuel *before* the severe cooling effect of the boiler takes place.

At this point we logically come to the next item of heat losses: i. e., radiation. In a brick-set externally fired boiler this loss may amount to from 3 to 5 per cent of the heat of the combustible and in certain unusual cases considerably more. It may be reduced by covering the boiler setting with an ample thickness of asbestos or other heat insulating material.

It may be further reduced by employing boilers of the internally fired variety, that is boilers whose furnaces are surrounded by water-heating surfaces, like the Scotch marine, the locomotive and Manning upright types. The loss due to radiation is of course much reduced in these boilers. But sometimes the lack of hot brickwork and insufficient combustion space results in such incomplete combustion, especially under forced conditions with soft coal, that the slight increase of efficiency by reduced radiation is more than offset by the loss occasioned by unburned fuel gases forced into contact with the cold boiler surfaces before their ignition and combustion can take place. In one test of this kind upon an internally fired boiler

I found at times enough unburned gas in the chimney to run a gas engine. It does not pay to go very far to reduce radiation by internal firing, especially when soft coal is burned. The advantages of the internally fired boilers are principally governed by local conditions such as limited floor space and absence of brickwork which will reduce upkeep expense.

The item of radiation may be roughly computed in the case of internally fired boilers by means of the area exposed, or covered with a material whose heat flow is known, together with the temperature of the steam and surrounding air. For brick-set boilers it may be obtained by subtraction of the other losses, but in most instances is of insufficient moment to demand the expenditure of great effort in its accurate determination.

We have now reviewed, one at a time, all the essential losses which constitute the entire waste connected with the operation of a steam boiler, and we have also had a practical example of the application of such an analysis to a factory boiler plant. In that particular instance an immediate saving of 25 per cent of the coal was the direct result. And when financial factors were combined with the thermal a means was discovered for

again reducing the fuel cost about 50 per cent more.

Thus we have an idea of the theory and practice of scientific investigation as applied to the boiler plant.

Chapter IV

THE BOILER PLANT (*Continued*)

A GREAT many questions arise in every plant regarding the economy of this or that contemplated change in operation or equipment. If such considerations were properly regarded at the beginning, before the plant was designed, there would not be the present large number of factory boiler plants that are operating with a large preventable waste.

In one plant for instance I found excellent boilers and a good type of stoker furnace, but poor efficiency. Here the trouble was due to lack of investigation beforehand. The boilers were not balanced to the load conditions and the furnaces were too big for the boilers. A boiler could not be taken off for cleaning without running the other much beyond its economic rating, which caused a great chimney loss. If both boilers were operated, the furnaces were so large for this appor-

tionment of load that they had three times as much grate surface as should have been provided for this condition, and this resulted also in a heavy loss of fuel. This was a difficult case to improve, since the great mistake had been made of buying equipment out of proportion to the work to be done. One of the first considerations in the design of a plant is the question of the number and size of the boilers to be specified. In deciding this matter three special requirements must be met.

1.—Extra boiler for cleaning.

2.—Efficiency at different factory loads.

3.—Growth of plant.

The efficiency of a boiler at different rates of driving is of primary importance in the above determinations, and before discussing them further it will be of advantage to understand what is meant by the ''rating'' of a boiler, and what factors influence its horse power or capacity. The mistake is frequently made of basing the purchase of a boiler on a simple horse-power rating. That this practice is most unsafe and results in much trouble will be quickly realized when I state that boiler manufacturers differently define a rated horse power as related to heating surface. There is, however, an accepted standard, that of the American Society of Mechan-

ical Engineers, which states that a boiler horse power shall be considered the evaporation of 34.5 pounds of water per hour into steam from and at 212 degrees. This is equivalent to the evaporation of 30 pounds of water from a feed-water temperature of 100 degrees into steam at 70-pounds pressure, and is roughly the amount of steam required to produce a horse-power hour with a simple Corliss engine at this pressure and non-condensing. Thirty pounds of water per hour is therefore a boiler horse power.

Now the voluminous data on boiler testing in the past have indicated that an ordinary boiler operates at its best efficiency when 2½ to 3 pounds of water per hour are evaporated for each square foot of its heating surface[1]. At these rates the development of a horse power would require

$$\frac{30}{2\frac{1}{2}} \text{ or } \frac{30}{3} = 12 \text{ or } 10 \text{ square feet}$$

of heating surface respectively. Consequently a 100-horsepower boiler should contain 1,200 or at least 1,000 square feet of heating surface. But 30 pounds of water per hour *can* be evaporated with as little as 5 square feet, whereas it was comparatively recent practice to allow 15. Here, then, is a

[1] Heating surface—That portion of the metal in a boiler which has water on one surface and fire or hot gases on the other.

range of 300 per cent in the factor determining how much heating surface makes a boiler horse power.

Consequently a boiler containing 1,000 square feet may be rated at 100 horse power at 10 square feet, or 200 horse power at 5 square feet, or only 66 horse power at 15 square feet of heating surface per horse power; so if these factors are not stated a rated boiler horse power means nothing.

I had a case where the plant owners had believed that horse power was horse power, and that was all there was to it. They purchased two boilers guaranteed to give an overload of 50 per cent. As there was no specification regarding the amount of heating surface, the manufacturer supplied only 7½ square feet per horse power instead of 10 or 12 square feet, which should have been called for in proper specifications. Now when the boilers were installed and operated under the guaranteed overload of 50 per cent, a horse power had to be developed with 5 square feet of heating surface. The result was that the boilers were actually run at 100 per cent over their normal rating, and their thermal efficiency was so low and the coal bill so high that it was necessary to call in expert advice, all of which might have been avoided if the plant owners had in the first place

taken steps to learn what a boiler horse power meant before attempting to use the term in the equipment of their plant.

The amount of heating surface that can economically develop a boiler horse power is at the present time a subject of close study and experiment and depends upon a knowledge of the laws of heat transference. With present-day equipment it is wise to provide at least 10 square feet of heating surface, although we may look for a considerable reduction of this figure in future practice. In any event, the capacity of a boiler will always be proportional to its heating surface and a purchaser who does not demand a full allowance is cutting himself off from a part of the capacity and efficiency to which he is entitled.

In a boiler test it is always important to determine at what per cent of their true rating the boilers are being driven, as this seriously affects their economy. Their true rated capacity is first obtained from a measurement of their heating surface in square feet, which divided by 10 results in rated horse power. Then if a boiler plant is found to be operating at considerably less than its rated capacity, and if the furnaces are correctly proportioned to the boilers, it is altogether likely that a large "chimney loss"

will be found. For this condition would indicate that the grate surface is too large for the rate of coal consumption, and consequently a surplus of air supply to the fires may be expected. This is simply stated by assuming that under a given set of conditions a square foot of grate will supply enough air for the proper combustion of 20 pounds of coal per hour. Then if only 10 pounds are burned, the grate may supply double the amount of air required for combustion, with the attendant waste described under "chimney loss".

Having now touched upon the meaning of capacity, we can refer with better understanding to the matter of fixing correctly the number and size of boilers adapted to best efficiency under any given set of local conditions. Perhaps the clearest way to indicate the method of determination of this matter will be to quote from one of my reports the portion bearing directly on this question, connected with the particular requirements which are stated or implied. While this plant was a small one the treatment illustrated governs all cases.

The first consideration in the design of any plant is the load to be carried. In the summer time as per test made on boilers this average summer load was found to be 162 boiler horse power. Allowance

should be made for an increase in the use of steam in the winter time owing to an additional load of about 300 amperes as compared to 800 in the summer time. This makes an increase of 37.5 per cent, so that the maximum average load for winter at present may be calculated at 223 boiler horse power.

Consequently the boiler plant should be able to handle 162 horse power to 223 horse power economically, and also allow for cleaning a boiler when necessary, and in addition to these requirements must be so designed as to provide for an increase in power requirements with a growth of the business.

Now if one extra boiler be provided for cleaning purposes, you can have either two boilers or three boilers.

If two boilers are selected, then one boiler alone must carry at good efficiency a load ranging from 162 to 223 horse power, and allowing 50 per cent for growth of plant it should also be able to handle 335 horse power. This is too high a load for a high-pressure horizontal tubular boiler unless it is of special design.

If three boilers are specified, then two of these boilers must carry 162 to 335 horse power. If they are each 150 horse power normal rating then one boiler will handle the 162 horse-power load very efficiently, and two boilers will handle the 335 horse-power load with equal efficiency, which includes the allowance for 50 per cent growth over present maximum-load requirements.

The intermediate maximum winter load of 223 horse power could be handled by one boiler at 148 per cent of its normal rating, or by two boilers at 74 per cent of their combined rating.

Three boilers therefore make the best number for economical running under these various load requirements, and would take care of the following specifications:

1—Extra boiler for cleaning

2—Economy at different loads
3—Growth of plant.

Now if the power requirements should increase from the present maximum winter load of 223 horse power to 446 horse power, this proposed plant of three 150 horse-power boilers would handle the load (which includes a 100 per cent increase of power) very efficiently with all three boilers running. If one boiler were taken off for cleaning it would mean taking 446 horse power out of two 150 horse-power boilers, or running two boilers for a day at 148 per cent of their capacity. This could be done very easily with fairly good efficiency.

Therefore three 150 horse-power boilers would really provide for 100 per cent increase over the present maximum power requirements, and would make a plant adapted to handle all loads between the present light summer load and a possible future load of twice the present maximum requirements, and all loads in between, at high efficiency.

Each boiler should have 1,800 square feet of heating surface, that is 12 square feet per horse power.

Very efficient operation is possible at reduced capacities. I have obtained over 71 per cent efficiency on a 200 horse-power boiler (rated at 12 square feet per horse power) when it was developing slightly over 100 horse power. In this case the grate was very small, which permitted a normal rate of combustion without an excessive air supply, and the high efficiency was also attributable to a specially controlled furnace and draft.

If a boiler is forced beyond its best normal

rating it is not able efficiently to absorb the large amount of heat developed by the combustion of the fuel. Consequently this surplus heat passes the boiler and appears as additional chimney loss.

With an allowance of 12 square feet of heating surface per horse power, a boiler with a well designed furnace may be driven with good efficiency at 50 per cent above its normal rating. This corresponds to 8 square feet of heating surface per boiler horse power developed. In some cases it may pay to increase this amount of forcing to double the rated horse power. This is done for instance to meet peak loads in the 59th street Interborough Station, New York City, where cost of space and investment for additional boilers would be more expensive than the loss of efficiency occasioned during the few hours per day that this measure is necessary. This would seldom if ever be advisable, however, in a factory power plant. For the usual factory imposes a comparatively steady load on the boilers, so that their number and size may be specified to give efficient operation, providing the load requirements are studied in advance.

A question often involved in factory power investigations relates to the comparative fuel economy of fire-tube *versus* water-tube boil-

ers. I have found that there is a general feeling that water-tube boilers are inherently more efficient than the fire-tube type. But this is not true. The brick setting of the water-tube boiler presents the greater radiating surface and consequently it is at a slight disadvantage in this respect. Otherwise there is no intrinsic reason why better economy of fuel should be expected with one kind than the other, providing of course the furnace is equally well designed in either case.

The real factors governing the selection of type are—steam pressure, load curve, facility of cleaning, first cost, and sometimes floor space in the case of large units.

With a comparatively steady load such as obtains in most industrial plants, and with an engine plant which does not demand a steam pressure above 140 pounds, the horizontal tubular boiler would show a marked advantage over the water-tube class providing the plant were not of such large capacity as to require too great a number of the fire-tube boilers. This advantage would be apparent in greater facility of cleaning, lower first cost, and settings generally better adapted to the combustion of soft coal. If the load to be carried is subject to severe fluctuations and sudden heavy demands for steam, the

water-tube boiler must be specified since it contains less water and consequently is more quickly responsive to forcing of the fires for sudden and rapid steaming. That is why this type is almost invariably chosen for public-service plants where the load with its tremendous "peaks" demands an immediate response from the boilers. Then again in this service, where production of cheap power is the sole object without the opportunity to cheapen it still further by the utilization of exhaust steam, only high-pressure engines and turbines are employed. This requirement necessitates the specification of water-tube boilers since the other type as a matter of design and strength is not adapted to high pressures. Then, as before implied, where large units are advisable the water-tube has the advantage, for the reason that the safe pressure of a tube or cylinder is inversely proportional to its diameter. Therefore when the size of a fire-tube boiler has reached a definite point, it becomes impracticable to develop larger units without disproportionate expense unless it is desired to meet very special conditions. An internally fired tubular boiler of the Scotch type, where the products of combustion do not come in contact with the shell, may however be made in larger sizes than the horizontal fire-tube type, and

will therefore frequently compete with water-tube boilers as largely evidenced in marine practice.

The cost of evaporation is an incidental though useful determination in plant investigations. This is commonly expressed in "cost of fuel for evaporating 1,000 pounds of steam from and at 212 degrees". It is obtained directly from the observations of our boiler test, and is most useful as a figure of commercial comparison of one plant with another. Depending upon the fuel used and the location and efficiency of the plant, this item will vary between 10 cents and 28 cents in this country. Its use facilitates the determination of the total cost of power and heating when combined with the other essential factors involved in that result. Given a fixed price and heat value of coal, the cost of evaporation will be inversely proportional to the combined boiler and furnace efficiency, and in a plant where the above values are known to be constant this term can profitably be used as an accurate measure of gain or loss in its operation.

A consideration of the condition of the feed water, both as to its temperature and as to its scale-forming properties, is essential to a complete investigation. It is needless to state that the accumulation of scale on

boiler heating surfaces resists the transfer of heat and therefore acts to reduce the efficiency of steam production. Also, owing to this resistance to heat flow, the boiler metal between it and the fire, being unable to transfer its heat to the water, will retain it and consequently become so hot as to cause bagging or blistering of the iron, resulting in a serious and sometimes dangerous injury to the boiler. The consumption of additional fuel due to scale is manifested in the production of an abnormally high chimney loss. Various figures are available which claim to show what percentage of the coal is wasted because of different thicknesses of scale, but I do not know of any truly reliable data of this description. It is also likely that different kinds of scale have different heat-resisting properties, which consideration is generally disregarded. Furthermore, unless comparative tests, made for the determination of such information, were in each case conducted with combustion of exactly the same quality and with a consumption of the same amount of coal, the resulting efficiencies would in no wise indicate the effect produced solely by the scale. It would also be difficult to find a boiler evenly coated with scale of anywhere near uniform thickness throughout. At present, therefore, it is sufficient

to desire to eliminate scale, as much from the standpoint of safety as for reasons of fuel economy, though both are important. Water contains two kinds of scale-forming "hardness": temporary and permanent. The temporary hardness consists principally of the carbonates of lime and magnesia and can be largely eliminated by raising the water to a temperature of about 190 degrees in an open vessel to allow the escape of carbonic-acid gas. An open type of feed-water heater with ample depositing and filtering arrangements is often the simplest solution. For neutralization of the permanent hardness, implying the presence of sulphates in the water, it is usual to resort to chemical purification. Special apparatus are available which are capable of combining the two operations for both kinds of hardness. Permanent hardness may also be reduced by duplicating a part of the action of the boiler itself, by the use of a closed purifier in which the feed water is subjected to a high temperature under pressure. These are questions involving chemical determination together with the best advice obtainable.

Referring now to the consideration of the *temperature* of the feed water, we are able to measure exactly what effect this will produce on our consumption of fuel. In the first

place, let it be clearly understood that the temperature of the water that goes to the boilers has practically no effect upon their efficiency in spite of the fact that we shall save money by previous heating of this water. Now if we recall our definition of efficiency it will be simple to understand this apparent paradox. Efficiency = Output ÷ Input. The output is the heat in the steam *above the feed temperature.* Since the boilers have no part in raising the water to the feed temperature, they cannot properly be credited with that part of the work. The combined efficiency of boiler and furnace is naturally that fraction of the heat of the coal which has been absorbed by the boiler. Therefore since the heat already in the feed water is not supplied by the coal, except as a reclaimed and independently variable factor, it must play no part in our efficiency calculations. And here is another fact that may strike us oddly. A boiler would show a slightly better theoretical efficiency on cold feed water. Again the reason is simple, because the greater the difference in temperature between the fire and the water in the boiler the more rapid will be the rate of heat transference, which will tend slightly to raise the efficiency in the case of using cold feed water.

Proper heating of the water supply to the boilers reduces the work imposed on them and consequently reduces the amount of coal consumed in like proportion, the combined efficiency of boiler and furnace remaining practically the same as with the colder water.

This gain of over-all efficiency is calculable by comparing the heat added to the water with the heat necessary to generate steam at the existing steam pressure and the temperature of feed.

For instance, if exhaust steam is utilized in a heater to raise the boiler water from say 60 degrees to 210 degrees each pound of water receives 150 B.t.u. Now if boiler pressure is 125 pounds per square inch, the heat required to make a pound of this steam from the old temperature of 60 degrees was 1,164 B.t.u. We have therefore saved or reclaimed 150 ÷ 1,164 = 12.9 per cent of the heat formerly required for the making of our steam, and this saving is directly reflected in a correspondingly great reduction in coal consumption. Approximately one per cent of the coal is saved for every 11 degrees we raise the feed water, providing this is accomplished with heat otherwise wasted. The matter of exhaust-steam heating of the boiler feed is always included in a plant examination, and frequently a substantial improve-

ment is found possible in this direction, owing usually to bad design in the plant itself.

It is possible and also advisable when conditions are favorable to raise the feed water to a much higher temperature still by utilizing or reclaiming some of the chimney loss from the boiler. This is accomplished by means of an economizer, consisting of a series of pipes installed in the path of the chimney gases, the feed water being pumped through the pipes in the opposite direction to the flow of the gases for promoting the most efficient transfer of the heat. This of course involves an expensive apparatus and its net value must be determined separately for each individual case. The water by this means may reach a temperature of 250 to 300 degrees before entering the boilers, but as the efficiency of the economizer increases, the temperature of the chimney gases is reduced. This together with the frictional resistance imposed upon the draft reduces the effectiveness of the chimney, and frequently to such an extent as to require the installation of mechanical-draft apparatus, which of course adds to the expense and complication of the plant. All of these factors must be carefully included in our calculation for ultimate commercial efficiency before a recommendation of real value can be formed.

In the case of one client, for instance, I found that comparatively simple changes for the improvement of the combustion of the coal would give as great a saving as would have been possible with the more expensive apparatus. Frequently the improvement of the furnaces will reduce the chimney loss to such an extent that an economizer would have very little heat to reclaim.

There are other instances where economizers are plainly indicated by careful investigation. Where it is for local reasons advisable to drive the boilers very hard, the economizer may result in a double saving that will be well worth while as a large paying investment. Not only is a great part of the chimney loss reclaimed as a direct saving of fuel, but at the same time the duty imposed upon the boilers is radically reduced, thus bringing about two decided economies.

The efficient handling of coal and ash is often an important matter. A careful study is made to determine exactly the amount of labor that could be saved by the installation of a proper conveying system, with all deductions made from the apparent saving to cover interest, maintenance, repairs, and labor connected with the project. The first cost and comparative efficiencies of different systems are taken into this consideration. In

ash handling there are two principal methods employed: the mechanical conveyor and the air conveyor. In the latter a swift current of air is produced in a tube, either by means of a fan or by the employment of steam jets acting to induce the required velocity. There are many cases where a good investment can be found in these directions.

It has been intended in these pages to set forth the methods employed for the determination of preventable losses, and by illustration from working cases to indicate to some slight extent the directness and effectiveness of these methods. It is quite aside from our present purpose to enter upon a discussion or even a fair description of the material aids in the production of the increased efficiency which is ever the goal of our efforts. The omission of a detailed treatment of equipment includes a very particular object. It is desirable for the purpose in hand to concentrate our attention upon those underlying principles which govern the economy of every boiler plant in a manner entirely apart from, and absolutely independent of, the specific nature of its equipment or kind of operation to which this apparatus is subjected. With this in mind we have now seen how each loss that may occur in a boiler plant can be traced to its source and accurately

measured. We have seen how it is thus possible with such a complete diagnosis to learn precisely what is wrong in the economy of any plant, and that the proper means for the saving of the preventable losses are indicated with equal clearness to the trained investigator.

After an industrial boiler plant has been subjected to a thorough investigation, and after the subsequent changes in operation or in equipment have been made for the elimination of waste, it then becomes necessary to render permanent this high efficiency and to prevent the plant from falling back into old ways and bad habits. This measure is particularly necessary in view of the fact that efficiency usually depends as much upon intelligent management as upon the excellence of equipment. This is especially true in the boiler plant, where bad management may reverse the savings of good apparatus into inefficiencies and continual waste.

We shall therefore consider the means used to maintain high efficiency. A simple system is installed which results in accurate daily records of the amount of coal burned, the weight of water evaporated, the temperature of feed water, and the pressure of steam. The last two figures are best obtained by means of automatic recording temperature

and pressure gages. The coal and water may be weighed, preferably by automatic scales and recorders. Many different styles of water meters are on the market and a large number of these are worse than useless. But there are also a few excellent water meters and weighers which are practicable and reliable.

A report blank is written up every night by the chief engineer, a copy of which is sent to the office, and this sheet shows the boiler horse power developed and the "equivalent evaporation per pound of coal as fired", together with the number of boilers under steam, the other data previously mentioned, and any remarks by the chief engineer which may have direct bearing upon the efficiency that was obtained. By this record any falling off of results is at once brought to the attention of the management, and calls for an immediate inquiry into the cause. After the equipment has once been made right, the cause of inefficiency can always be found either in the coal itself or in the handling of the boilers and furnaces. If an examination of the coal does not show a falling off in heat units or an increase in moisture of sufficient extent to account for the decreased evaporation on the report, then it is certain that the boiler-room staff is responsible for

the waste that has been exposed. Such defective operation is usually attributable to careless or ignorant firing, but may be partly chargeable to dirty boilers, leaky settings that have not been kept in repair, improper draft regulation, etc.

Such a system can be further elaborated to such degree as local conditions and size of plant may warrant. Under appropriate circumstances an automatic combustion recorder will prove a valuable adjunct to the daily record sheet. This device produces a continuous record of the CO_2 in the flue gases and therefore acts as an excellent check upon the handling of the fires for correct supply of air.

With a daily system in its simplest form it will always be known whether the right number of boilers are in operation for the load that is carried. The lack of this knowledge alone leads many plants into excessively heavy coal bills.

The form reproduced in Fig. 9 shows a daily report blank which I devised to suit the local requirements of one of my clients and it will serve as an illustration.

The installation of an accurate accounting system for the boiler plant operates for the instruction of the engineer and fireman and enlists their interest in the work of produc-

ing economy such as no other means will accomplish. Both humanity and efficiency are well served when a system of rewards is inaugurated to compensate the boiler-room staff for developing and maintaining a high evaporation as recorded on the daily re-

DAILY BOILER REPORT
C. F. SMITH & CO. NEW YORK CITY, N. Y.
DATE

	Day 7 A.M.-6 P.M.	Night 6 P.M.-7 A.M.
Total Coal .... A		
Parts of soft coal		
Water Evaporated .... B		
Actual evaporation per lb. of coal (B ÷ A) C		
Temp. of Feed Water .... D		
Steam Pressure		
* Factor of Evaporation .... E		
*Equiv. Evap. from and at 212° (C x E) = F per lb coal .... F		
* Average B. H. P. Developed, Approx.	B ÷ 330	B ÷ 390
Number of boilers fired		
Head Fireman		
Number of boiler cleaned to-day if any		
How found		

CO_2 — Chemists special report on combustion—

Remarks and Suggestions:- Engineer is requested to used this space. Quality of coal or any cause for poor results, repairs or trouble of any kind to be reported

(Signed,

Engineer,

* These columns signify calculations which can be made in the office. but the engineer should check them.

The Engineering Magazine

FIG. 9. FORM FOR DAILY BOILER REPORT

ports.[1] Such reward is well deserved by the men, and can not easily be neglected by the management since the method reacts with large profits to their advantage in the reduced coal bills.

We have now seen that the efficiency of a factory boiler plant is definitely determin-

[1] See Evaporation Standards in Chapter X.

able by means of scientific diagnosis, that the balance of energy known as "loss" can be measured and analyzed, and that the large preventable portion of that loss can be reclaimed and added to the efficiency. As emphasized by our foregoing discussion, we have seen that for these accomplishments we are dependent upon the universal laws of the conservation of energy.

So far we have dealt with the transformation of the latent chemical energy of the coal into the active heat energy of combustion, which was again transmitted by the agency of the boiler into heat in the form of steam. From this point we proceed to the transference of this heat *to* the engine, and then its action *in* the engine by which a part of it is converted into mechanical work, thus representing another cycle of efficiency. The investigation of these matters, together with their bearing upon the location and elimination of preventable losses as applied to the factory plant, will form the subject matter of our next discussion.

Chapter V

STEAM PIPING AND THE ENGINE PLANT

FROM the boilers the steam is piped to engines for power, to pumps and auxiliary apparatus, and to systems for heating and process work. For the present we shall confine our discussion to that portion supplied directly to the engines.

There is always a certain loss of heat from the steam on its way from the boiler to the point of its application. This loss is caused by radiation of heat from the steam piping and by energy used to overcome the friction of the steam against the inside surface of the pipe. The combined loss from both causes is manifested in condensation of a part of the steam and a reduction of pressure at the end of the pipe.

In a well-designed plant these losses are very small. The condensation caused by radiation of heat to the surrounding air is minimized by properly covering the pipe with

layers of heat-insulating material. In still air the radiation from a bare pipe with high-pressure steam may be computed by allowing a loss for each square foot of surface of from 2 to 3 B.t.u. per hour per degree difference of temperature between the air and the steam in the pipe. This has been determined by experiment, but varies with conditions. The heat so radiated is not lost except in so far as our engine is concerned, but it is misspent energy used in raising the temperature of the air which envelops the pipe. Another method of determining the extent of this loss is by trapping and measuring the condensation in the steam pipe and adding to this the entrained moisture in the steam, which is found by means of a steam calorimeter.

The friction loss may be approximated by a comparison of the amount of heat in the steam at the initial and final pressures. A good covering will prevent about 85 per cent of the radiation loss.

The reduction in pressure caused by friction is lessened by shortening the pipe, increasing its diameter, and eliminating all possible elbows and turns.

The total combined piping loss between the boilers and the engines may be very small, so that the heat delivered to the engine may be 98.3 per cent of the heat in the

steam produced by the boilers, which figure would represent the efficiency of the steam pipe correspondinng to our illustration in Chapter II.

In some cases where long lines of piping are used and are badly laid out or improperly covered, the loss will be great; and in one plant that I investigated the preventable portion of this waste amounted to about $3,000 worth of coal a year.

In addition to correct design and covering of piping it is possible to increase the saving by the employment of superheated steam. There are three kinds of steam. 1, Wet steam, which contains moisture in the vesicular state—that is, water in the form of fog. 2, ''Dry saturated'' steam, which contains no moisture and has the temperature exactly corresponding to its pressure; this the layman calls ''ordinary dry steam''. 3, Superheated steam, which not only contains no moisture but carries more heat than would be accounted for by its pressure. It therefore has a temperature above that of dry saturated steam, but the same pressure. Superheated steam is obtained by passing ''ordinary'' steam through specially constructed coils or tubes, surrounded by hot gases either from the boiler furnace or from a superheater having a separate furnace of its own.

By such means it is common practice, where conditions warrant, to add from 100 to 200 degrees to the temperature of the steam after it leaves the boiler. For special requirements very much higher temperatures can be added.

The addition of heat to steam under pressure requires the expenditure of about 0.55 B.t.u.[1] for each degree per pound of steam providing the steam had no moisture at the beginning, in which case the heat required is correspondingly greater since this moisture has also to be evaporated with the attendant input of its latent heat at the existing pressure.

In the case of piping steam to distant points, that part which reaches its destination as water is wholly worthless for doing work in the steam engine, although it has caused the consumption of as much fuel per pound as the uncondensed and therefore useful portion. Now by first passing our steam through a superheater, which in the given instance will absorb from the coal 55 heat units per pound of steam, we shall add 100 degrees to its temperature. Since it is a property of steam that it will not condense until it becomes "dry saturated"—that is,

[1] The specific heat of superheated steam varies with the pressure of the steam. Fifty-five heat units added to steam at 105-pounds gage pressure would add 100 degrees of superheat.

until it has lost its superheat—we may convey it in pipes and avoid condensation as long as any of the superheat remains. In good practice a degree of superheat will carry the steam 10 feet, so that if we desire to convey our steam 1,000 feet we should have to add 100 degrees to its temperature in order that its entire weight may be available for use at the end of the line.

The additional consumption of coal for securing the superheat must of course be deducted from the total saving of steam. This amount with a steam-gage pressure of 101.3 pounds will be $(55 \div 1{,}161^{1}) = 4.7$ per cent of the heat required for a pound of the "dry saturated" steam. In the case of superheaters which are arranged inside of the boiler setting, less coal than this would be required for the reason that the additional heating surface would tend to increase the efficiency of the combined boiler and superheater and to produce a smaller chimney loss. Consequently in ordinary superheating a very small percentage of fuel is required, and the saving by the stoppage of condensation in long pipe-lines may be very large, depending entirely upon local conditions.

Assuming now for the progress of our

[1] Heat in a pound of dry saturated steam at 101.3 pounds gage or 116 pounds absolute above 60 degrees.

argument (as in Chapter II) that the pipe-line to our engine has an efficiency of 98.3 per cent, and (as before stated) that the boiler and furnace supplied to this line 58 per cent of the heat of the coal, then the engine will receive in the form of steam $(0.983 \times 0.58) = 57$ per cent of the original heat in the coal.

We are at present considering a fair type of factory Corliss engine running non-condensing, and we have seen in our previous statement of analysis that of the heat it receives only about 7.3 per cent is converted into mechanical energy or work at the belt wheel or jack shaft.

Now it might be assumed that so great a loss in this conversion from heat to work is principally chargeable to an inefficient design of engine, but we shall see that this is not entirely true by comparing its efficiency with that of the most economical steam prime movers in existence. Take for example the highest type of steam turbines or compound engines operating on high-pressure superheated steam and we find a maximum efficiency of about 24 per cent. At the present writing a 25,000 kilowatt steam-turbine electric-generating set for one of the Chicago Public Service stations is being constructed, and this machine will be guaranteed to deliver a kilowatt hour of electric current for

11.25 pounds of steam with an output of 20,000 kilowatts. It will operate on 200 pounds pressure and 200 degrees of superheat, and an absolute back pressure of 1 inch of mercury. Assuming mechanical and electrical losses of 7½ per cent, a brake horse-power hour at the shaft will be produced for about 7.76 pounds of steam. If the boilers produce 7.76 pounds of steam under actual conditions per pound of coal at the turbine nozzles, then a mechanical horse-power hour will be obtained from one pound of coal, apparently a wonderful economy. But let us see just what this means in true efficiency. The number of heat units in a pound of the steam as supplied is 1,310 B.t.u., so that the total heat energy delivered per horse-power hour is 10,165 B.t.u. Now since a horse-power hour is equivalent to 2,545 B.t.u. at 100 per cent efficiency, the actual efficiency developed is 2,545 ÷ 10,165 or 25 per cent. This represents about the greatest achievement of the kind in modern steam-power development, and yet we have succeeded in converting into useful work only 25 per cent of the heat in the steam. By a little analysis we shall see how much further we are likely to attain toward higher efficiency. After the almost complete expansion of the steam in the highly developed turbine (or engine as the case may

be) it enters the condenser[1], at which point it still contains 885 B.t.u. per pound. This large amount of heat is necessarily taken up by the circulating water of the condenser and is therefore worthless for the purpose of producing work. Therefore all the heat in the steam that could possibly be converted into mechanical energy with an absolutely perfect engine using this kind of steam would be 1,310 — 885, or 425 B.t.u., which is 425 ÷ 1,310 = 32.4 per cent of the total heat in the steam. This represents the ultimate efficiency of an ideal engine or turbine working under these almost ideal conditions. We may understand how nearly this limit has been approached by comparing the actual efficiency of 25 per cent that was obtained with what the perfect prime mover would develop, viz:—32.4 per cent, and we see that the actual efficiency is 25 ÷ 32.4, or about 77.2 per cent of a perfect result. That is to say, no matter how we strive to improve the efficiency of turbines or engines, we could with a perfect machine utilize only 32.4 per cent of the heat of the steam (at 200-pounds gage pressure and 200 degrees superheat) and we

[1] Vacuum created by condenser assumed at 96.6 per cent of a theoretically perfect vacuum, i. e., an absolute pressure of 0.505 pounds per square inch or a 29-inch vacuum. Atmospheric pressure is 14.7 pounds per square inch at sea level.

shall have already reached 25 per cent with the new turbine we have mentioned.

This narrow margin of theoretically possible improvement is imposed by the fact that no matter how much we may improve our prime mover, the greatest portion of the heat of the steam (even when expanded to almost a perfect vacuum) will still be necessarily discharged to the condenser. This part of the heat is latent and is given up only upon condensation.

Still higher pressures and temperatures of steam call for very expensive and special design of boilers, piping, and apparatus and the gain that can be looked for in these directions is too small to make the attendant savings of any considerable commercial value at the present cost of fuel.

It is now plain that however much we improve the steam engine or turbine, its thermal efficiency will be low, the reason being that whether we operate condensing or non-condensing, with compound cylinders, multistage turbines, or with a simple Corliss engine, the greater portion of the heat (that is, the latent heat of steam at the exhaust conditions) will pass out to be absorbed by the condenser or to appear in the exhaust steam from the engine.

It is now easy to understand why the effi-

cient utilization of exhaust steam is of primary importance in the factory plant. It is also easy to see that a moderate-sized factory power plant so arranged, designed, and operated as to balance the production of exhaust steam to the heating load, can develop an over-all efficiency much higher than the average large and highly refined plant of the public-service corporation. A failure to accomplish this balance in such a manner as to operate efficiently at different seasons of the year will often make the cost of power so high that the central-station or water-power development will be enabled to offer prices for power which are competitive. Success in this matter depends upon a full knowledge of local conditions, obtainable only by such careful investigation as we are gradually describing; and this knowledge together with correct engineering information must form the basis of all plans and recommendations.

An important question always to be decided is whether it will pay to run the engine condensing as opposed to exhausting into a heating system. If there happens to be a use for all of the exhaust steam all of the time this becomes a simple problem to solve. For illustration, take the Corliss engine of our example. If we add a condenser to pro-

duce the ordinary vacuum of 25 to 26 inches we shall expect to reduce our 30 pounds of steam per brake horse-power hour to about 22½ pounds, thus making a saving of about 25 per cent [1] of the steam and coal formerly used to run the engine. We shall also increase the horse-power capacity of the engine by the percentage which the vacuum adds to the mean effective or driving pressure on the piston. This frequently amounts to one-third more power. For exact calculations of these results each case must be separately considered and proper allowance made for all affecting conditions. For the present purpose it is sufficient to state that under usual *factory* conditions the saving by condensing over non-condensing will be in the neighborhood of 20 to 25 per cent of the steam required per horse-power hour.

[1] For condensing—The heat in dry saturated steam expanding adiabatically from 101.3 pounds gage or 116 pounds absolute to 2 pounds absolute, i. e., running condensing (which corresponds to 25.85 inches of vacuum, is 265 B.t.u. The corresponding heat between the same initial pressure and 5.3 pounds back pressure or 20 pounds absolute is 132 B.t.u. That is, if *perfectly utilized*, steam of 101.3-pounds gage pressure will perform a trifle over twice the work, if by adding a condenser the back pressure is reduced from 5.3 gage pressure to a vacuum of 25.85 inches of mercury.

There are cases where this ratio of improvement has been realized by adding a condenser and inserting a low-pressure turbine between the engine and condenser, a turbine being intrinsically more efficient on low-pressure steam than a reciprocating engine. But the addition of a condenser alone to our simple engine will actually give an increase of power output of only about one-third, or a saving of steam for the same power of twenty-five per cent. These figures will vary considerably with the conditions such as initial steam pressure, mean effective pressure originally obtained, design of valve gearing, size of ports, condenser connections, etc.

If now we refer back to the heat in the exhaust steam from our Corliss engine and remember that it contained 92.7 [1] per cent of the heat in the steam supplied to it, we shall observe that the expedient of employing a condenser will save in the form of energy only 20 to 25 per cent, and the balance of 72.7 per cent to 67.7 per cent will still be wasted as discharge [2] from the condenser. Add to this the fact that an appreciable percentage of steam is required to operate the condenser and the over-all efficiency will be still lower.

If, on the other hand, we are able to connect the engine exhaust with an efficient heating system, we shall be able to displace expensive live steam from the boilers and utilize practically all of the heat from the exhaust steam. A good feed-water heater should form the first part of such a system. Now to complete our comparison of condensing *versus* heating with exhaust steam, let us see just what efficiency the engine is capable of developing in the latter instance.

We shall assume that the heating system and feed-water heater are so well designed

[1] 88.1 per cent "dry saturated" steam and 4.6 per cent moisture at 228 degrees.

[2] A small percentage of this can be returned to the boiler by adding a few degrees to the boiler water by means of a feed-water heater placed between the engine exhaust and the condenser.

that together they will utilize all the heat in the exhaust steam down to a drip temperature of 200 degrees, and without for the present considering any piping loss, which may be very small or of considerable importance as governed by local conditions.

The heat in a pound of exhaust steam at 5.3-pounds back-pressure that it will give up when cooling to 200 degrees at the drip end of heaters and radiators is 940 B.t.u.[1] This is 940 ÷ 1,161 = 80.8 per cent of the heat entering the engine[2] or 940 ÷ 1,076 = 87.3 per cent of the heat in the exhaust steam, including its moisture. Of the heat entering the engine 7.3 per cent was converted into mechanical work, and by the perfect utilization of the exhaust steam we convert 80.8 per cent more of its heat into useful purposes (live steam of equal heat value would otherwise be required) so that the true efficiency of our ordinary Corliss engine has become 7.3 per cent + 80.8 per cent = 88.1 per cent. There is no prime mover in the world that will even approach this efficiency, which is

[1] A pound of exhaust steam at 5.3 pounds gage pressure containing 5 per cent moisture cooling to 200 degrees atmospheric pressure. In the

steam 0.95 (1,156 — 168) =	938.6 B.t.u.
In the water 0.05 (228 — 200) =	1.4
Total heat given up per lb. of exhaust steam	940.0 B.t.u.

[2] Assuming dry saturated steam at the engine and computing heat above 60 degrees.

obtained as we have seen by balancing the production of exhaust to the heating requirements with a simple type of steam engine. When we considered running condensing for this engine we found that we saved not over about 25 per cent, which brought its efficiency from 7.3 to 9.75 per cent, whereas by the above method of combining power with heating our efficiency becomes over 88 per cent. In practice this works out so beautifully that for approximate calculations we may say that the efficient use of exhaust steam will take the place of an equivalent amount of live steam direct from the boilers. Thus we have seen a proper comparison of the value of running condensing as compared to heating with exhaust steam when the heating load just balances the production of the by-product exhaust steam. So also have we confirmed the statement previously made that the average factory plant has a great intrinsic advantage over the central power-station where the former requires a considerable amount of heating. The figures for this comparison may be represented by a ratio as great as 88 to 24, that is almost 4 to 1. When there is a smaller proportion of heating to be provided for, this ratio varies accordingly and must be worked out carefully for any given case. As to condensing, the rough statement

that it will not pay if more than one-third of the engine exhaust can be used is approximately correct.

If, under the operating conditions cited, it is desired to know the cost of producing energy or horse-power hours alone, the efficiency of the engine must be credited with the heat it supplies to the heating system. This is equivalent in this case to crediting the engine with a (thermal) efficiency of 88 per cent.

This 88 per cent efficiency of a non-condensing engine exhausting into a perfect heating system as above stated is comparable to the 24 per cent efficiency of the highly developed condensing steam turbine of the great and efficient central power plants. And this is the main reason why an isolated plant is able to compete successfully with power produced by the large central steam or water-power stations.

This discussion strikingly exposes the necessity of including in our power-plant investigation a determination of the amount of steam used for heating as well as the amount consumed by the engines. These tests should include the power *versus* heating requirements for summer and winter, day and night, in order to design for high efficiency at all times and under all conditions.

Where such determinations were made in one case I was enabled to add about 50 per cent to the horse-power capacity of the plant with only an insignificant increase of steam from the boilers. In this instance this was accomplished by converting a large amount of live-steam radiation into a low-pressure system which permitted the substitution of exhaust steam. The latter was then supplied by a Corliss engine which was installed to "take the work" out of the high-pressure steam before it passed to the heating system. The engine merely acted as a reducing valve and the power obtained was a by-product of the revised heating system.

There are many buildings and factories today that make steam for heating and purchase electric current for their power purposes but might just as easily obtain most of the power they need as a pure by-product of the heating system, with very little further expenditure for fuel; yet this fact is not widely understood outside of the engineering profession. The exact extent to which such results can be accomplished cannot be prophesied or estimated without a very careful preliminary investigation of all affecting conditions along the lines we have indicated.

In order to gain full information of these conditions it is necessary to test the engines

for steam economy. It will then also be known whether steam is being wasted by the engine in the form of exhaust that might be utilized, or in case of there being no opportunity for this, to determine the saving possible by reducing the waste in the engine itself. Even if the exhaust steam from one or more of the engines perfectly balances the heating load it is quite possible to discover waste and to cut down the consumption of steam of those engines in the plant which are run condensing. Economies of this kind react directly on the coal bill.

The steam consumed per horse-power or kilowatt hour is found by measuring the feed water of the boilers supplying steam to an engine and at the same time taking indicator cards and noting the kilowatt hours produced. In case of an engine exhausting into a surface type of condenser the exhaust condensation can be measured instead of the feed water if this happens to be easier under the circumstances. From these tests we shall have the steam and coal[1] consumption in the engine as well as the fuel cost per horse-power or kilowatt hour, and we shall know how efficiently the engine is working,

[1] Coal consumption of the engine is obtained by combining with its steam consumption the evaporative results of the boiler tests discussed in Chapter III.

how much steam it is wasting, what percentage of its rated horse power it is developing, and consequently just what saving can be effected in steam and coal by making such repairs, changes in load, steam pressure or back pressure, etc., as will have been indicated by the results and data of our test.

Sometimes a heavy back pressure [1] is found which when relieved by proper changes will increase the capacity of the engine 20 to 50 per cent and also reduce the steam consumption materially by allowing an earlier cut-off. It is quite usual to find enough back pressure.[2] on a non-condensing engine to reduce its horse-power rating from 12 to 25 per cent, and in one case I found, owing to heavy back pressure, sufficient steam in the exhaust to run twice the number of engines for the steam that was used. Trouble from this source is found by examination of indicator cards taken as part of the engine test. It is corrected usually by improvement in the exhaust piping or heating system, which is

[1] Back pressure is the pressure of the exhaust steam acting on the opposite side of the piston from the live steam that drives the engine.

[2] Effect of back pressure is proportional to the ratio of back pressure to mean effective pressure as found by the indicator. Thus if the mean effective pressure is 40 pounds and the back pressure 8 pounds, the capacity of the engine by relieving the back pressure will be increased 8 ÷ 40 or 20 per cent. For the same horse power this will permit an earlier cut-off in the engine with the attendant saving of steam. The effect of reducing back pressure is the same in kind as adding a condenser, the increase of efficiency varying only in degree.

frequently so poorly designed as to cause the congestion of steam resulting in back pressure on the engine. I have even found serious back pressure caused by the sticking of the relief valve. This was remedied in five minutes and would never have occurred if a proper back-pressure gage had been attached to the exhaust pipe to indicate the existence of trouble. Sometimes back pressure is found to be caused in the engine itself due to its design in the provision of inadequate exhaust passages. The effect of this is especially apparent when running at full capacity or overloads.

If none of the exhaust steam from an engine can be used for heating purposes it then becomes the duty of the investigator to determine the extent and cost of savings that can be effected by changes in operation or by replacing the engine with one of more economical design. Since the test has given complete data as to the load to be carried and also as to the efficiency of the engine, it becomes possible to predict how much saving would be obtained by a more highly developed type of engine whose economy is known. All items of cost, including any added operating charges, must be taken into account so that the net saving on the investment can be determined. Such a report must review all

types of engines and turbines that are applicable to the case in hand, and many important practical considerations must be included such as are connected with the operation of the factory, the hours of running, the class of help at hand, the space available, and the steam pressure and condensing facilities both existing and possible. The cost and saving connected with the use of superheated steam must enter into these computations, both as applied to the present engine and to its possible use with the improved unit under consideration. Our engine test has given us a record of the variation of the load to be carried at all hours of the working day and this factor has important bearing upon our selection of a new unit.

At this juncture let us emphasize the relation of "load factor" to power-plant efficiency. "Load factor" is that percentage of its full rated output which a plant actually develops during a given period of time. In central-station practice the time period is considered 24 hours, though the load factor may also be determined for a 12-hour day. If a station is designed to produce 500,000 kilowatt hours per day, but owing to the impossibility of obtaining steady loads its actual output is only 175,000 kilowatt hours, then its load factor is said to be 35 per cent,

a very common condition. Now it is obvious that when a plant so operates its investment charges per kilowatt hour will be almost trebled, and this indeed is the most serious problem with which the great power stations of today have to contend. Public service imposes a widely variable load and includes such high "peaks" at certain hours of the day that the load factor is necessarily low. These variations in power requirements not only affect the investment charges, but impose an uneconomical kind of load on the engine or turbine equipment.

To a large extent the average factory is free from the more serious part of such trouble, especially in regard to the load to be carried during the working day. There are exceptions to this rule, but in any event it is important to understand each case for the determination of best results, and our all-day test has given us such information that we can select the best sizes of engine for the work. An engine gives its highest efficiency at full load, falling off either with an increase or decrease from this point of the horse-power output. This relation of efficiency to capacity is generally clearly shown by a curve plotted with reference to a series of parallel determinations for the two values. This is known as the efficiency

curve and the characteristics of any kind of engine or turbine may be readily compared by this means which frequently facilitates a decision upon the best unit for the load conditions which latter have been disclosed as indicated.

In our final specification of prime mover we must embody a due consideration for increased horse power to allow for growth of the factory plant. But this must be obtained with a minimum sacrifice of efficiency at other than full loads. To add somewhat of definiteness to this statement, we may, for instance, by the use of reliable efficiency curves, be enabled to discover that for some given size of unit a turbine will give better efficiency under a wide variation of load than a compound engine, although it may also be found that at a steady full load the engine would have the advantage.

Without further discussion we shall realize that the thorough investigation of the engine-room part of our problem involves both broad and at the same time specific information, which is the result of decades of study and progress in this field. The subject is so great and so involved in its analysis that it would be quite aside from our present object to do more than make a few simplified

statements in a rather comprehensive manner.

We shall therefore leave to the following chapter the review of efficiencies obtained with steam engines and the comparison with economies obtainable from other prime movers.

Chapter VI

STEAM PIPING AND THE ENGINE PLANT
(Continued)

LET us review the range of efficiencies obtained with steam engines, upon which we have already touched, and concentrate our attention upon the preventable losses involved, together with a comparison with those of other types of prime movers which enter the field of competition and which consequently demand our consideration. These other types include the internal-combustion engine whose fuels are oils and gases, the latter when derived directly from coal giving the name of "producer gas" to that specific kind of apparatus. To these we may also add the water-wheel or turbine.

Beginning with the steam engine we have already observed that its efficiency varies between [1] and 25 per cent, considered purely

[1] The old slide-valve type will have efficiencies as low as 4 per cent. The coal used per engine horse power varies of course with the efficiency of the engine, the efficiency of the boiler and furnace, and the quality of the coal. In wide ranges of practice an engine will consume from about 1 or 1½ pounds of coal to 9 or 10 pounds per horse-power hour.

as a power producer without reference just now to the utilization of its valuable by-product of exhaust steam. We have seen that even an ideal engine or turbine would not give an efficiency above 32.4 per cent with steam at 200-pounds pressure and 200 degrees superheat, and that the efficiency is limited by the practicable amount to which we can increase the pressure and temperature of the initial steam, these in turn being limited by considerations of design, safety and cost. The large necessary loss with the steam engine or turbine is occasioned by the discharge to the condenser of the exhaust, which even at highest obtainable vacuums contains the majority of heat units originally carried in the steam supplied to the prime mover. The principal partially preventable losses in the process are those due to friction, cylinder condensation, leakage, incomplete expansion of the steam, "wire drawing", and excessive clearance volume.

The losses due to friction in the best types, depending on their rating, amount to only about 5 per cent of the total power produced, which is a proportionately smaller percentage when referred to the heat or energy in the steam. The percentage of frictional loss decreases as the horse-power output increases. It is reduced by careful design with

special regard to balanced valves and proper lubricating systems.

The loss due to cylinder condensation is the most serious of all, and is susceptible to the greatest reduction by improved design. It is caused by the chilling effect of the cylinder walls of the engine upon the entering steam, and in simple engines using dry-saturated steam frequently accounts for 20 to 25 per cent of their steam consumption. There are three methods of importance employed for reducing this loss:

First, by increasing the horse-power rating of an engine, which is accomplished by the use of higher speeds and pressures and later "cut-offs". The percentage loss thus becomes less since the condensation is an approximately constant quantity.

Second, by the use of superheated steam, which must be cooled to its saturation point before condensation can take place, it is made to act more nearly like a perfect gas with a consequently lower consumption by the minimizing of condensation. Under the right conditions the amount of coal required to superheat the steam is very small compared to the saving of steam thereby produced in the engine. This practice is much older abroad than in the United States, but is well established in this country and rapidly gain-

ing favor in manufacturing establishments. Economies are so decided in this direction that all modern central plants are employing superheat. Factory plants are able to obtain greater saving by its use than the more highly developed central stations. The reason is twofold. Factories usually require longer steam lines and therefore there is a good saving possible in the reduction of piping loss. But the more potent fact is that the less efficient the engine the greater the cylinder condensation, and consequently the greater will be the economy to be gained by superheating the steam which acts directly to reduce this loss. Steam of very high temperature requires the use of special design and material for piping and engine valves, but experience has indicated that a total temperature of not over 450 degrees to 500 degrees will not seriously affect an ordinarily well-constructed factory power plant.

The third method of reducing the largely preventable cylinder condensation is by compounding our engine or turbine; that is to say, instead of performing the complete expansion of the steam in one stage or cylinder, distributing the work consecutively among two or more cylinders of progressive size. Thus we have the compound engine (double, triple, or quadruple) according to the number

of stages of expansion which we provide. This step is advisable only when comparatively high-pressure steam is available with its consequent high temperatures. The transfer of heat from steam to metal is proportional to the difference in temperature between the steam and the metal. Hence when this difference is very great it pays to divide the expansion of the steam by compounding the cylinders so that each cylinder is kept at a temperature corresponding as nearly as possible to the temperature of the steam in that particular stage of its expansion. In this way cylinder condensation is reduced and a higher efficiency is obtained. It follows also that in such an engine superheating the steam will have less effect than on a simpler type of steam motor. In fact, it is often possible to get compound-engine efficiency by combining superheat with a simple engine, especially if the steam pressure is not high, that is not over about 125 pounds.

In a number of cases it has been found that a low-pressure turbine can be attached to the exhaust of a steam engine with a large increase in horse power and efficiency. This is simply a convenient and efficient method of compounding and amounts to the addition of a second or third cylinder to the engine.

Before leaving the subject of loss due to

cylinder condensation let us emphasize an important point in this connection. Other things being equal, the larger the engine or turbine the greater will be its efficiency and the lower its steam consumption per horse-power hour. This follows from causes already discussed, and permits appreciation of the fact that economy is produced by using a few large engines rather than a larger number of small ones. In factory work it is frequently possible to secure large savings by following this plan. Otherwise stated, the development of mechanical energy by steam engines or turbines should be concentrated.

This concentration as a general rule is limited only by load conditions. For example, if a plant were to develop only one-third of its full load for a large portion of the time it would not ordinarily pay to try to operate with a single large engine. Either two or three engines suited to the conditions and thrown on as the load required would give better efficiency. On the other hand, if all the exhaust steam can be used all of the time, then it makes but little difference whether by scientific design we produce an engine horse-power hour for less steam. In such a case I have used a reducing valve for a Corliss engine when it had to run at very light loads. This enabled the "chief" to

increase his steam pressure as the load increased. Without such precaution the engine valves would have "knocked", and "loops" or negative work would have occurred in the cylinder accompanied by regulation troubles.

The engine problem can never be properly considered without reference to the use of its exhaust steam.

In another factory plant I found the main engine to be about four times too large for the maximum load. In order to get it to run smoothly they had reduced the boiler pressure by 30 pounds. The result of both causes was a steam consumption of 80 pounds per horse power per hour and most of the exhaust was being *wasted.* In this case it was necessary to recommend a smaller engine which would give an efficiency of 30 pounds, permit the use of full boiler pressure, and pay for itself in a short time. The great loss was due principally to cylinder condensation, most of which would be re-evaporated and wasted through the exhaust relief valve.

The loss due to leakage is an important one. This occurs through the steam valve or past the piston, so that live steam escapes through the engine into the exhaust piping without doing any work. This kind of waste is very serious and I have found losses of

many tons of coal from this source. The correction is usually a machine-shop job involving reboring of the cylinder, new piston rings, new piston, or replaning of valve and seat according to the precise location of the trouble. In a large measure it can be prevented in the original specification of engines by avoiding those types, especially most designs of high-speed four-valve or automatic engines, which are intrinsically wasteful in regard to leakage after they have been run a few months. In a recent test I found an engine of this class using 57 pounds of steam per horse-power hour where 30 pounds should have been sufficient. The principal part of this loss was due to leakage past the steam valve.

The loss from incomplete expansion of the steam may be caused by the lack of intelligent specifications in the purchase of the engine. That is to say, many engines are purchased ''on faith'' with no proper provision as to their size, speed, cut-off at full load, or steam consumption. Thus it is possible to buy an engine guaranteed to develop say 200 horse power. This may easily result in the purchase of an engine which should properly be rated at 125 horse power. It will develop 200 horse power, but only at the expense of a late cut-off which means an excessive steam

consumption. Unfortunately this is not an uncommon occurrence. The waste of steam is due to incomplete expansion caused by insufficient cylinder volume.

The excessive overloading of an engine *properly* purchased also prevents complete expansion and results in waste. This occurs in the rapid growth of factory power requirements the extent of which is not realized by the management.

Wire drawing is the term applied to the reduction in pressure occasioned by the passage of steam through a restricted port. By its action the initial steam pressure, acting on the piston of the engine, becomes less than the pressure at the throttle. The effective pressure of the working stroke is thus reduced, with a corresponding increase in the steam consumed per horse-power hour. This loss should be reduced to a minimum by the designer of the engine as very little can afterwards be accomplished for its correction.

The last in our list of preventable losses is that due to excessive clearance volumes in the cylinder and steam passages. When the engine is on dead centre the space enclosed by the piston, the cylinder head, and the steam and exhaust valves is called the "clearance". Evidently this space must be

filled with steam before full pressure can be exerted on the piston. The live steam so used is wasted as compared to the operation of an ideal engine which would have no clearance space. By scientific design the clearance is made as small a percentage of the total cylinder volume as possible, and in operation a high "compression" may be carried in order to fill the clearance space with exhaust instead of live steam.

Now if the investigation of our plant has shown that none of the methods thus far discussed will yield the desired results in lower cost of power, we can still turn to other means.

The internal-combustion engine includes oil, producer gas and natural gas as fuel, which in each case is burned inside of the engine cylinder. The heat thus developed causes pressure which acts in place of steam against the piston.

If we already have a steam plant a part or all of which must be run to meet imposed requirements, then the addition of *another type* of power unit will add complications which are well to avoid unless great economy demands them. This is especially true in *manufacturing* where the *product* and not power is the first consideration. Therefore in most cases it does not pay to employ two

kinds of power. Such a change would usually add labor and interest charges far out of proportion to those required for an equal increase in power by a steam-driven unit. Furthermore the internal-combustion engine, although it has undergone great improvement in recent years, lacks the decided element of reliability that can rightly be attributed to the steam engine.

But there are still many cases where the gas engine is an economic proposition. Notably we may mention those manufacturing localities where cheap natural gas is available, where I have tested many engines giving excellent service, both as to reliability and economy. Then there is a growing field of usefulness where the gas engine is employed to operate on the by-product gases from the blast furnaces and coke ovens of the steel industry.

The producer-gas engine is applicable in small plants requiring very little steam and where an occasional shut-down or a duplicate engine would be considered.

The oil engine has the advantage over producer gas in that the engine itself constitutes the entire plant and does not require an additional apparatus for producing the gas.

The cost of power with any of the internal-

combustion engines depends upon the same factors that govern any other means of power production, viz:—Efficiency, cost of fuel, labor and interest charges. The last two factors must be determined separately for any individual set of conditions. Efficiency and cost of fuel we shall now examine.

In practice, depending upon load and other conditions, the Diesel type of internal-combustion engine when operating on oil fuel will develop an efficiency of 25 to 27 per cent. Under best conditions about 31 per cent has been obtained. Thus if the heat value of the oil used was 20,000 heat units (B.t.u.) per pound, the engine used 0.41 [1] pounds of oil for each brake horse-power hour generated. Other highly developed oil engines attain these efficiencies. The oil consumption will vary with the percentage of full load carried, but with a well-designed high-pressure four-cycle engine 0.6 pounds of oil of the above heat value may be taken as a fair estimate of practical performance. If the fuel costs 2½ cents per gallon containing 7½ pounds, then the cost of oil per brake horse-power hour will be $0.002 [2], that is two mills.

Fuel used = x
Then x (0.31 × 20,000) = 2,545
x = 0.411.

[2] $\$0.025 \times \frac{0.6}{7.5} = \$0.002$

This is not necessarily lower than the fuel cost with a steam engine using coal, for the comparison depends entirely upon efficiencies and fuel prices. A brief comparison may not be amiss at this point. With low-grade coal like anthracite buckwheat, a properly designed boiler plant will give 1,000 pounds of steam at a fuel cost of 11 cents. (I have a very small plant doing this for 10 cents.) A good compound engine or turbine will give a horse-power hour for 15 pounds of steam, which at this price gives a fuel cost per horse-power hour of $0.00165, that is *1.65 mills,* which is even lower than the oil-engine fuel cost just cited. There are plenty of manufacturing plants where the cost of fuel per horse-power hour will run from 5 mills to 1½ cents, and great savings are generally possible, but by methods that can be indicated with certainty only by a thorough investigation of *all* the surrounding conditions.

The efficiencies of the best types of internal-combustion engines designed to operate on the Otto or four-stroke cycle [1] will range between 16 and 24 per cent. When they are

[1] Four strokes of the piston or two revolutions for each explosion or power stroke. Order of cycles: 1, explosion stroke outward; 2, piston returns to dead centre expelling gases of combustion to exhaust; 3, suction stroke outward taking in fresh charge of gas and air; and 4, compressing same preparatory to firing for the first or power stroke. In the two-cycle engines all four functions are performed in two strokes with an explosion every revolution.

supplied with gas from a "producer" their individual efficiencies remain about the same, but the over-all or ultimate efficiency is reduced by the loss in the producer itself. The producer efficiency may be said to be about 80 per cent, so that if the engine has an efficiency of 20 per cent the ultimate efficiency in the use of the fuel will be 0.20×0.80 or 16 per cent, based on the heat in the coal supplied to the producer.

In the natural gas regions I have found gas engines using from 10 to 12 cubic feet of gas per indicator horse power per hour with gas of 1,100 B.t.u. per cubic foot selling at 20 cents per 1,000 cubic feet. In these circumstances an engine may develop a brake or mechanical horse-power hour for 11 cubic feet of gas, in which case the fuel cost [1] would be about \$0.0022. (The heat efficiency of the engine would be 21 per cent.) This is a fairly low figure in manufacturing work, but it must be remembered that in many localities where gas is cheap high-grade coal may also be obtained at very low prices. Furthermore, in the many cases where a considerable amount of heating is required, the steam en-

[1] Fuel cost $= \dfrac{11 \times \$0.20}{1{,}000} = \$0.0022$ per b.h.p. hour.

Assuming 90 per cent mechanical efficiency, i. e., 10 per cent friction, a consumption of 10 cu. ft. per indicator horse-power hour will become $1.00 \div 0.90 \times 10 = 11.1$ cu. ft. per. *brake* horse-power hour.

gine by the utilization of its valuable exhaust steam may not only be enabled to compete with the gas engine but may produce a much higher over-all efficiency.

It is true that the gas or oil engine also possesses a similar feature of by-product heat supply, but not to the extent of the factory type of steam engine. Referring back to our previous analysis we shall remember that the exhaust steam from a good Corliss engine may contain over 90 per cent of the original heat in the steam supplied at the throttle. By its efficient utilization its exhaust may be made to take the place of nearly an equal weight of live steam from the boilers, in which circumstance the over-all efficiency thus produced is extremely high.

In order to understand the nature of the by-product heat from our gas engine we may profitably consider the typical heat balance on page 141. This analysis may fairly be taken as representative of the best gas-engine practice and we shall take for the example the engine recently mentioned.

An examination of this analysis at once shows where the principal losses occur in the operation of a gas engine. The greatest loss and the most tangible one is the heat carried away in the jacket-water used for the purpose of maintaining the cylinder at

GAS-ENGINE HEAT BALANCE

Heat in 1 cubic foot of natural gas.	1,100 B.t.u.
Heat supplied by 11 cubic feet of gas to produce a brake horse power per hour—11 × 1,100....	12,100 B.t.u.
Heat equivalent of 1 brake-horse-power hour....................	2,545 B.t.u.
Heat efficiency of engine 2,545 ÷ 12,100 =.....................	21 per cent

	B.t.u.	Per cent
Heat supplied to engine per brake-horse-power hour in 11 cubic feet of gas.............	12,100	100
Heat converted into mechanical or useful energy (efficiency)...	2,545	21.0
Heat absorbed by jacket water..	5,082	42.0
Heat absorbed in friction with a mechanical efficiency of 90 per cent. 2,545 B.t.u. ÷ 0.90 × 0.10 = 283.................	283	2.3
Heat discharged in hot exhaust gases and losses due to radiation and incomplete combustion.......................	4,190	34.7
Total..................	12,100	100.0

a safe working temperature. This cooling water may enter the cylinder at a probable temperature of 60 degrees, and in practice is drawn off at about 100 to 120 degrees. It is better, however, for gas consumption to maintain a higher jacket temperature. I have increased the horse power of a factory gas engine 11 per cent by maintaining a tem-

perature between 170 and 180 degrees. The practicability of this measure depends largely upon the design of the engine, although the deposit of impurities from the water might in some cases cause trouble. In experimenting on another make of engine no increase of power was gained, for the evident reason that in that particular case the gas inlet valve also became heated, thus increasing the specific volume of the gas so that the cylinder received a less weight of fuel per stroke, and so offsetting the gain that could otherwise have been expected.

Furthermore, in factory work it is usual to provide an over-abundant supply of cooling water to prevent possible trouble. This could be managed by automatic regulation of the cooling water, but no great advance has been made in this direction. The warm jacket water, even at the usual moderate temperatures, may however be used for heating purposes and this is sometimes done. But since it is about 100 degrees below the temperature of exhaust steam, a relatively large heating surface must be employed to obtain good efficiency. With a properly worked-out heating system a considerable fraction of the jacket-water loss may be reclaimed and credited to the gas-engine's efficiency if conditions admit of such a plan.

The remaining large waste is the last item in the preceding analysis. By far the greatest portion of this, in all probably 30 per cent, may be accounted for by the heat carried out in the exhaust gases. This heat may be partially conserved by the use of a heater or low-pressure boiler through which the engine exhaust is made to circulate, thus giving up heat to raise the temperature of the contained water or to produce a limited amount of low-pressure steam which may be utilized in any convenient manner. Various proposed plans have included hot-water or steam heating, and even the propulsion of a condensing steam turbine by means of the steam resulting from the hot exhaust gases. But this matter of heat reclamation as pertaining to the internal-combustion engine is neither so simple nor so widely practised as in the case of the steam engine whose exhaust is so readily and effectively applied to heating purposes. Practically all the waste heat from the steam engine is discharged through the exhaust pipe and is contained in *one form only,* whereas the gas engine wastes its heat both through its exhaust pipe in the inconvenient form of hot gases, and also through its cylinder walls, appearing in the form of warm water usually of insufficient

temperature to be especially suitable for heating purposes.

"Producer gas" is made from coal supplied to a special retort or producer in which a moderate temperature is maintained. The volatile gases of the coal are first expelled and collected in a gas tank. The remaining incandescent carbon is then treated with steam to form water gas, which mingles with the volatile or "coal gas" and the mixture thus formed is known as "producer gas".[1]

The process of making this gas is not perfect. The losses are principally due to radiation of heat, unused carbon discharged in the ash, the presence in the gas of surplus oxygen, and the formation of CO_2. The result is that the gas produced contains only about 80 per cent of the heat value of the coal used for its production. Consequently an engine of 20 per cent efficiency using gas from such a producer will give an over-all efficiency of $0.20 \times 0.80 = 16$ per cent, as before stated. That is, 16 per cent of the heat in the coal will have been converted into useful mechanical work and the total loss will have been 84 per cent.

[1] Producer Gas: The volatile portion of the coal is made up of hydrocarbon gases, largely CH_4 and similar groups. The water gas consists of CO and H_2, the reaction being $C + H_2O = CO + H_2$. A certain unavoidable percentage of CO_2 and O dilutes the gases and reduces their heating value.

A producer-gas plant that develops an actual horse power per hour for one pound of coal under operating conditions is doing very fair work.[1]

The fuel economy of the producer-gas plant has had a very desirable effect on designers and manufacturers of steam equipment. The Germans have made great progress in the development of small-sized steam plants which have been given the name of "Locomobiles". These are now manufactured in the United States. Such an equipment consists of a well-designed engine mounted directly on top of an internally fired boiler with provision for highly superheating the steam. To reduce cylinder condensation the engine cylinders are continually bathed in the hot chimney gases discharged from the boiler. There is practically no steam piping between engine and boiler, and every detail is so arranged as to concentrate and conserve to the greatest possible extent the heat involved. The result is that a horse power per hour has been produced for approximately one pound of coal, thus practically equalling the producer-gas equipment in fuel economy.

In the large refined central station of re-

[1] If the coal used has a calorific value of 13,000 B.t.u. per pound, the over-all efficiency in this case is 2,545 ÷ 13,000 = 19.6 per cent.

cent design it is also possible to generate a horse-power hour of energy for close to a pound of good coal, so that when the most highly developed types of each are considered the internal-combustion engine cannot show any marked saving in fuel over the economy of the steam engine. In factory work the question of heating is important, and in this field the steam engine is highly adaptable. But each problem must be separately analyzed and decided in accordance with its local conditions, and first cost and reliability must be considered.

Before leaving the engine-room department of our investigation the reader would perhaps feel disappointed if some reference were not made to the present rivalry which in a certain way exists between the reciprocating steam engine and the steam turbine. A few years ago there was a prevalent opinion, based on the performances then obtained, that in general the reciprocating engine was the more efficient for high steam pressures and that the turbine gave the better results on low pressures. It was also an opinion of the times that in sizes under about 1,000 kilowatts the engine would give higher efficiency than the turbine. While it is still true that there are intrinsic reasons why the turbine is more efficient on low pressures, yet

the turbine has made such rapid improvement in design during the last five to seven years that its position relative to the reciprocating engine has considerably altered. It is now possible to equal engine efficiency in turbines of 500-kilowatts capacity and even considerably less. In the very small sizes, i. e., from 30 to 100 kilowatts, especially non-condensing, the engine has the advantage in steam economy. Owing to the continual advance in steam turbines, improvements in reducing gears, smaller floor-space requirements, absence of oil in exhaust steam, etc., it is not always possible to decide a case of engine *versus* turbine on a small difference in fuel consumption alone, especially in the moderate sizes. Again it is necessary to refer back to local conditions of operation. For instance, if all our exhaust steam were to be utilized in any case, a few pounds greater consumption on the part of the turbine would make no economic difference. On the other hand, if coal were dear, condensing impracticable, and sufficient heating requirements were unavailable for utilization of the exhaust steam (considering moderate or small size of unit) the steam engine would win the case, other things being equal. Since conditions of operation including steam pressure, superheat, condensing facilities, size of unit,

cost of fuel, heating requirements, etc., so largely govern the selection of prime mover, it is safer to make no further statements than those already indicated.

We have yet to consider the water turbine as a factor in our engine problem. In only limited localities are factories able to make direct use of the water turbine, the modern development of one type of the old-time mill wheel. In most cases, however, the water turbine has only to be considered indirectly as a means for the production of "outside power" for sale by electric power companies. Incidentally a modern water wheel or turbine ranges in efficiency from about 70 to 90 per cent. That is to say, in the latter case only 10 per cent of the energy in the falling water would be misspent or wasted. In many sections of the country the question of the value of purchasing power from an outside source as opposed to the production of power in the industrial plant itself constitutes a problem of growing importance. Such outside power may be generated hydraulically, or by water power combined with an auxiliary steam plant, or by a steam plant alone. The method of determining its value to the factory owner will be the same in any of these cases, providing the factor of reliability is always equal.

Local factory conditions so vitally affect the cost at which power may be produced that without an individual investigation it is impossible to state whether a plant would reduce or increase its expenses by purchasing outside power at any proposed rate. The two most important of these factors are: 1, possible use for exhaust steam for heating or process work as compared to steam required for power production; and 2, cost and quality of coal available.

Knowledge of the first factor must be thorough. It must be determined how much exhaust steam under improved conditions can be used at all seasons of the year for both the day and night runs. The amount of live steam demanded by the engine must be similarly determined. Only by this means will it be possible to learn the true average cost of power. The necessity of this measure will be understood by reference to our foregoing discussion on the value of exhaust steam for heating as compared to live steam.

In order to demonstrate this relation to the owners I once shut down all the engines in a large plant and continued the heating with live steam. (The exhaust steam from the engines had previously supplied the heating system.) They were astounded to find that the same quantity of fuel was burned

with the engines all stopped and the boilers carrying the heating load only. If outside power had been installed for the machinery no saving in coal would have resulted.

When the heating accomplished by the exhaust steam is deducted from the fuel required to operate the engines, the *balance only* is chargeable to *power,* and such is the method required in order to obtain a true and fair comparison of the cost of operation with purchased power for running the machinery and with boilers reserved for heating.

A certain part and sometimes all of the boiler plant will be needed for heating even when outside power is used. The fixed and operating charges on this portion of the boiler plant are therefore directly chargeable to the heating system and not to the engine plant in estimating power costs. When the true cost of power per kilowatt hour (on the switchboard) has been worked out for conditions as found, it then becomes necessary to determine what this cost *would* be if the plant were improved and all preventable waste reclaimed. This improved or *possible* power cost is based upon the complete data obtained in the boiler and engine-plant sections of our investigation thus far outlined, with due reference to the heating-system ex-

amination which will be discussed in the following chapter.

Assume now that the possible cost per kilowatt hour has been determined, including all charges for recommended changes. This cost is *not* directly comparable to the kilowatt-hour cost as quoted by the electric-power company. To the latter's figure must be added the charge per kilowatt hour of that part of the time of the factory engineer and assistants which is chargeable to power alone since their services are not eliminated by the use of outside power. This point is usually overlooked, and as it often makes a marked increase in the true cost of the purchased power it is not usually mentioned by its promoters. This subject of central-station current will be treated more fully in a later chapter.

The engine room of every factory should be equipped with a simple system of daily reports designed to meet the conditions of the plant. Duplicate copies covering the last twenty-four hours should be filed each morning in the office of the manager and in the records of the operating engineer. A sample of a report blank made for the requirements of a client is shown in Fig. 10. The results to be obtained by the use of such a system are:

1. The establishment of an intelligent and economic relation between the management and chief engineer.

DAILY ENGINE ROOM REPORT

DATE

	Day 7 A.M.-6 P.M.	Night 6 P.M.—7 A.M.
Main engine hours running		
Small engine hours running		
K. W. H. Main Engine		
K. W. H. Small Engine		
Total K. W. H.		
*Coal per K. W. H.		
Maximum K. V. A. during run		
Heating Live		
Heating Exhaust		

Remarks: – Engineer is requested to use this space for reporting any matter needing attention.

Signed,

Engineer.

* The main engine should use not over 4.3-lb. coal per K.W. H. at 7-lb. actual evaporation and the small engine 6.4-lb. coal, surplus being chargeable to live steam heating and to pumps.

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Fig. 10. Form for Daily Engine-Room Report.

2. A working knowledge of the plant that would not otherwise be obtained, including a means for estimating the cost of power.

3. The computation of the efficiency of the

engine plant (the boiler plant being already under a system described in Chapter III).

4. Where the engines have been individually tested, the amount of live steam required for heating and process work may be calculated.

This record blank is in use in an electrically operated plant, but it can be modified for belt-driven practice.

During the investigation of the plant the engines were individually tested for steam and fuel consumption per horse-power and kilowatt hour. The kilowatt hours are obtained daily, and this information combined with the evaporation record of the boiler plant enables the engineer to compute just what portion of the total steam produced is used for pumps and live-steam heating.

Chapter VII

THE HEATING SYSTEM

A GREAT waste of steam and fuel in manufacturing plants is directly chargeable to inefficient heating systems. To most lay minds the term "steam heating" affords merely a picture of radiators or pipe coils, having connections for steam and drip lines, with possibly a supplementary idea that if the condensed steam is returned to the boiler the efficiency will be high and that there the matter ends.

If it truly ended so simply and happily there would be no excuse for the present discussion. As a matter of fact, however, I have found heating systems using three pounds[1] of steam where one pound would have accomplished the same result. In other cases I have discovered heating arrangements intended for exhaust steam so badly designed that costly live steam had to be used

[1] The term "pounds" refers not to steam pressure but to *weight* of steam used.

instead, while more than enough exhaust steam to heat the entire plant was being wastefully blown out of the relief valve. It is from such cases as this that the uninitiated have formed the opinion (and it is hard to dislodge) that live or high-pressure steam is far superior to exhaust steam for heating purposes.

It was shown in Chapter IV that the exhaust steam from a Corliss engine working under ordinary factory conditions contained 92.7 per cent of the heat present in the live steam at 101.3-pounds gauge pressure, and that the engine absorbed only 7.3 per cent of the heat contained in the original steam. A more striking comparison is that between the heating values of high- and low-pressure steam without regard to the action of an engine. The following table gives the desired figures:—

Heat above 60 degrees in 1 pound of steam at 101.3-pounds gauge pressure	—1,161 B.t.u.
Heat above 60 degrees in 1 pound of steam at 5.3-pounds gauge pressure.	—1,128 B.t.u.
Heat above 60 degrees in 1 pound of steam at 0-pounds gauge pressure	—1,122 B.t.u.
Heat above 60 degrees in 1 pound of exhaust steam, including 5 per cent moisture at 5.3-pounds gauge pressure, from example of engine in Chapter IV	—1,076 B.t.u.

From this we may see that a pound of steam at zero gauge or atmospheric pressure contains within 3½ per cent as much heat as the same weight of steam at 101.3-pounds pressure.

A board of directors found it difficult to believe that exhaust steam from the engines could actually accomplish about 90 per cent as much heating as live steam direct from the boilers. They asked me if it were true that a producer-gas engine could produce a horse-power hour from a pound of coal. To this I answered "Yes." Their next question was, "How much coal per horse-power hour do our present (old slide-valve) engines require?" I stated this amount to be between 5 and 7 pounds. "Why then," they demanded, "can we not make our power with a gas producer and save 4 to 6 pounds of coal per horse-power hour?" I reminded them that all the exhaust steam was being used, and that even if they eliminated all the steam engines they would still have to burn *about 90 per cent as much fuel* to make sufficient live steam *for the heating requirements alone.* In addition to this there would be the coal consumed by the producer equipment, and no saving would result.

This did not appeal to them, so I planned a demonstration which proved to be convinc-

ing. (This was briefly referred to in the last chapter.) I shut down every engine in the plant and weighed the fuel to the boilers whose load was heating only. The result was that no perceptible difference in fuel consumption was noted, much to the surprise of the interested officials. Needless to say, exhaust steam was respectfully given its true monetary value after that occasion, and the old slide-valve engines held sway as long as there was abundant use for the discharge of their exhaust.

This illustration demonstrates the importance of providing a heating system that will make efficient use of exhaust steam, since each pound so used reduces the otherwise necessary boiler steam by nearly a like amount. In fact, cases have been reported where *more* fuel was required to heat a plant with the engines shut down than with the engines running under full load while heating the building with their exhaust steam. This may be explained (in case pressure is used) by a consequent increase in the heat discharged by the radiators, as will be later discussed. Under these circumstances, as far as fuel is concerned, *the power obtained is a pure by-product of the operation of heating.*

Heating efficiency is defined by the same

formula as that used in boiler and engine testing. *It is output divided by or compared to input.* The input is the total heat in the steam delivered to the system, and the output is the heat which is usefully applied.

For illustration, let us examine the case of a single radiator which receives steam at 1.3 pounds above atmospheric pressure, all the steam being condensed in the radiator, which discharges its water at atmospheric pressure and at a temperature of 211 degrees. We then have the following figures:

Input per pound of steam............1,152 B.t.u.[1]
Discharge per pound of steam........ 179 B.t.u.
Useful output, or radiation per pound of steam............................ 973 B.t.u.

Efficiency = Output ÷ Input = 973 ÷ 1,152 = 84.4 per cent.

Thus 84.4 per cent of the total heat in the steam has been converted into useful radiation. The drip contains the balance, that is, 15.6 per cent of the original input. Now if this drip could be returned to the boiler without loss of temperature, as compared to feeding the boiler with water at 32 degrees, we should reduce the coal bill about 15.6 per cent. In this event, should we credit to the heating system the heat made useful by its return to the boiler, our over-all heating efficiency

[1] These heat values are measured from 32 degrees F.

would approximate 100 per cent, barring losses in steam and drip piping.

This illustrates maximum heating efficiency for low-pressure systems operating on close to atmospheric pressure. Such a system is highly effective for exhaust-steam heating, since the horse-power and steam consumption of the engine are not materially affected by this low back-pressure.

If it were practicable to hold the hot water of condensation in the radiator until its temperature was further reduced the efficiency of the radiator would be increased. On the other hand, the heat in the returns to the boiler would be correspondingly reduced.

It is almost needless to state that all hot drips from a heating system should be returned to the boiler. The foregoing example illustrates the effect on the coal bill of returning the drips when *all* the boiler steam is directly applied for heating purposes. *If only a fraction of the total live steam produced is used for heating, then the fuel saving effected will be proportionally reduced.* Thus in the above case, if a condensing engine required one-half of the boiler steam, then the returns of all the drips of the heating system would amount to only 7.8 per cent, instead of 15.6 per cent, of the heat in

the original boiler steam, and a corresponding part of the coal.

The element of time enters importantly into such calculations. Thus in this last case, if heating is discontinued in the warm months, the return of the drips will affect the yearly coal consumption to a still less degree, and computations of savings must be made to suit the exact conditions of operation applying to the case under consideration.

A common error among plant owners is to ascribe undue economy to the heat that may be carried in water resulting from the condensation of steam. They conversely greatly *under-estimate* the heat carried in exhaust steam. As a matter of fact the heat in a pound of water at 212 degrees is only 180 British thermal units reckoned above the freeezing point; whereas in a pound of steam at *the same temperature* the heat units number 1,150. Hence if this steam is condensed in a radiator it gives out 970 heat units, and the drip will contain 180 heat units. Nevertheless I have found the management very anxious about correcting the loss due to a few escaping drips, while at the same time large quantities of uncondensed exhaust steam were blowing away almost unnoticed. Pound for pound, the waste which attracted the manager's attention was equal to less than

one-fifth of the heat being carried away in the disregarded exhaust steam.

This is a good thing to remember, and it will stand twice telling:—*Utilization of the exhaust steam saves five times as much coal as the return of an equal weight of hot drips to the boiler.*

I do not intend to reduce the proper emphasis on the value of conserving hot drips, but I do desire to present the various phases of the power problem in their true relation to each other; for only in the light of such an understanding will industrial power users eventually be enabled to place their plants on a basis of high economy.

The efficiency of *radiating surface* is quite another thing from the efficiency of the process of heating. The former use is an unfortunate application of the same term, and indicates the *rate* of heating rather than true efficiency. Thus if a radiator or heating coil is so constructed as to impart to the air or liquid say 426 B.t.u. per hour for each square foot of its surface, it is said to have a higher efficiency than one which gives out only 300 B.t.u. per square foot. This "rate efficiency" is used to determine how many square feet of heating surface will be required to warm a building or to perform a given *amount* of radiation. It has therefore no fixed rela-

tion to the *waste* or *conservation* of the heat in the steam.

A comparison has been made to show the relative heating values of high- and low-pressure steam. It was seen that steam at atmospheric pressure contains within $3\frac{1}{2}$ per cent as much heat as at 101.3-pounds pressure. There are, however, two factors which should be considered before attempting the substitution of low- for high-pressure steam. Other things being equal, more radiating surface is needed for the low-pressure steam. This is because the *rapidity* with which the heat is given off is proportional to the difference in temperatures existing between the steam in the radiator and the surrounding air or liquid. A square foot of direct heating surface will transmit to still air about 1.8[1] B.t.u. in an hour for each degree difference in temperature between the air and the steam when that difference is about 142 degrees.

Let us see the effect that would be pro-

[1] The coefficient of radiation 1.8 increases gradually as the temperature difference increases. Fair values for the above example are as follows:

Steam pressure, Gauge pounds	Temp. Steam	Temp. Air	Temp. Diff.	Coefficient
101.3	339	70	269	2.30
20	259	70	189	1.98
0	212	70	142	1.80

For a given temperature-difference the coefficient also varies with the design of the radiator or heating coil. One cause of this is the interference of nearby pipes or sections. See Carpenter's "Heating and Ventilating Buildings."

duced on the heating of a building from changing the pressure of steam in the radiators. Assuming the temperature of the air to be 70 degrees and selecting the correct coefficients from the footnote, we shall find that a square foot of surface will radiate in an hour:

255 heat units with	0	lb. steam pressure
374 heat units with	20	lb. steam pressure
618 heat units with	101.3	lb. steam pressure

From the above it would appear that with a given amount of radiating surface, steam at atmospheric pressure would give only about 68 per cent of the heating effect to be derived from steam at 20 pounds. In actual practice, however, I have substituted the lower for the higher pressure and at the same time increased the heating effect from the radiators. This apparent discrepancy between theory and practice may be explained by the fact that the former system was poorly designed. The circulation of the steam was imperfect and the distribution of heat to the various departments of the factory was not uniform, consequently the entire radiating surface was not effective.

When the low-pressure *exhaust* steam was substituted certain changes were recommended which so improved the circulation and

distribution that the entire heating surface became active. This change more than compensated for the former advantage of the higher steam pressure.

Formerly, in this case, steam from the boilers at 125 pounds was lowered to 20 pounds by means of a reducing valve for the heating system. In place of this reducing valve I substituted a 150 horse-power non-condensing Corliss engine, which expanded the steam down to atmospheric pressure before it was used for heating. The power thus produced (at least about 90 per cent of it) became a pure by-product of the steam heating. *This resulted in producing an engine horse-power hour for about 3 pounds of steam or 0.375 pounds of coal, a performance that cannot even be approached by the largest and most refined type of central power-station that it would be possible to conceive,* for an ideally perfect engine would not even approximate this figure. This economy could not have been effected, however, without a very careful design of the exhaust system.

The second factor referred to as demanding consideration previous to the substitution of low-pressure for high-pressure heating is the question of temperature.

There is no limit as to the *amount* of heating that can be accomplished by an adequate

supply of exhaust or low-pressure steam, but since its temperature is lower than that of steam under pressure there *is* a limit as to the *temperature* obtainable by its use. For all ordinary heating of buildings, however, exhaust steam will give as high a temperature to the air as may be desired, and cases of dry rooms may be quoted where a temperature of 150 to 170 degrees is maintained with exhaust steam. We may remind ourselves also that exhaust steam in a properly designed feed-water heater will bring the temperature of the water to within about 5 degrees (or less) of its own temperature. For instance, with the steam at atmospheric pressure and having a temperature of 212 degrees, we may expect to secure feed water at about 207 degrees.

Heat cannot flow from a colder to a warmer body. Consequently the degree of heating possible is limited by the temperature of the steam used. Thus steam at 2.3-pounds pressure (temperature 219 degrees) could not under perfect conditions produce a temperature of over 219 degrees in the surrounding air or liquid, whereas steam at 101.3-pounds pressure under ideal conditions would raise the surrounding medium to its own temperature of 339 degrees. There are therefore certain applications where it is necessary to

heat with high-pressure steam in order to obtain the higher range of temperatures.

Before proceeding, and in close connection with the above points, it may be well for some of us to clarify our minds in regard to the difference between temperature and heat. Temperature is the measure of the *intensity,* or degree of heat. The heat unit or B.t.u. measures the *quantity.* For example, we may have two different weights of water at the same temperature of, say, 200 degrees. Although the intensity of heat is the same in each case, if the one mass of water weighs 10 pounds and the other 20 pounds, the latter represents double the *amount* of heat of the former. Thus temperature alone is no measure of heat, any more than *pressure* is a measure of the *quantity* of steam generated in a boiler.

High-pressure steam heating in the factory is likely to be less efficient than low-pressure heating when the discharge of the radiator is controlled by the usual steam trap. Efficiency is here considered in the true sense. If the high-pressure drip is directly fed to the boilers by means of the return type of trap, then the high-pressure may be as efficient as the low-pressure system, providing air binding is eliminated. The objection to this method is that the boiler-feed water

is introduced by two separate means (the return trap and the feed pump) and this interferes with its correct and convenient measurement for the daily records that are so essential to the efficient operation of the boiler plant.

The object of a trap is to permit the discharge of water condensed from the steam and to prevent the escape of all uncondensed steam. An efficient trap also provides for the release of air from the radiator. If it fails in this last respect the circulation of the steam is blocked and a portion or all of the heating surface becomes ineffective. The make-up water supplied to the boiler contains air which is conveyed with the steam to the heating coils. The steam condenses and is discharged by the trap as water, but unless special provision is made the air accumulates in the radiator until the passage or entrance of the steam is prevented, thus choking the system. It is this action of badly designed systems that has led to the erroneous belief that to secure proper heating it is necessary to use pressure and to "blow" steam through the radiator uncondensed. Each pound of steam blown out in this manner represents a loss of 100 per cent of its available heating value, hence the wastefulness of this useless procedure. Of course its

practice *does* result in driving the air out of the system, but it does so at the expense of an excessively heavy consumption of steam and fuel, and it is quite unnecessary if the heating plant is correctly laid out and specified.

The reason why heating with steam at a higher pressure is less efficient than with steam at atmospheric pressure (or thereabout) is that in the former case a considerable quantity of uncondensed steam is discharged with the water of condensation. This statement applies to heating systems equipped with the usual types of bucket or float traps which allow the drips to return to a vented hot well or receiver from which they are pumped back to the boilers.

Even assuming that such a trap is in perfect working order this blowing out of live steam will occur. The water which accumulates in the trap has about the same temperature as the steam in the radiator to which it is connected. Assume for example that the steam has a pressure of 20 pounds. Its temperature will be about 259 degrees and the water discharged by the trap will have nearly the same temperature. Hence when it discharges to atmospheric pressure it will give off steam until it is cooled down to 212 degrees. In this case the amount of steam

"blown off" will be equivalent to 57[1] heat units, or about 5 per cent of the total heat in the steam supplied to the radiator. The actual weight of water evaporated is also approximately 5 per cent of the weight of the original steam. This re-evaporated steam finds relief through the first vent pipe that may be provided, and is therefore wasted.

In addition to this item an added and more serious loss occurs with high-pressure steam heating. This is the discharge of live steam through the trap which accompanies the water of condensation, and it is due to the inefficiency of the trap itself. I have found traps blowing through so much steam in this manner that the waste from this cause amounted to approximately as much heat as was required to heat the buildings. This was due principally to bad design of the traps, and partially perhaps to the worn condition of their mechanism.

The higher the steam pressure the greater

[1] Calculation of loss by re-evaporation:—

One pound steam at 20-pounds pressure contains 1,167 B.t.u. above 32 deg.
One pound water at 259 degrees contains .... 228 B.t.u. above 32 deg.
One pound water at 212 degrees contains .... 180 B.t.u. above 32 deg.
One pound steam at 212 degrees contains ....1,150 B.t.u. above 32 deg.

Let x = weight of water evaporated into steam at 212 degrees.

Then 1 — x = weight of remaining water at 212 degrees.

$$1{,}150\,x + 180\,(1 - x) = 228 \text{ and}$$

$$x = 0.0495 \text{ pounds steam at } 212 \text{ degrees.}$$

Hence waste is $0.0495 \times 1{,}150 = 57$ B.t.u. per pound of steam fed to radiator.

This is $57 \div 1{,}167$, or about 0.049 of the heat in the steam supplied.

will be the losses caused by re-evaporation in the discharge and by the direct "blowing through" of uncondensed live steam.

A low-pressure system of heating is therefore more likely to be economical, entirely aside from the question of exhaust steam. But it is necessary to point out at this juncture that the economy of low-pressure heating is *not* effected *at the boiler.* I have found superintendents and managers who had an idea that if they could change a boiler from high- to low-pressure they would produce a direct saving in fuel. Let us remember, as before stated, that it requires within 3½ per cent as much heat to make steam at 0 pounds gauge pressure as at 101.3 pounds, and the fuel required in either case is very closely proportional to these values. To be strictly accurate, the boiler operating at the lower pressure would have a lower temperature, so that there would be about 127 degrees greater difference between the boiler and furnace temperatures, which would tend toward a slightly better absorption of heat from the fire. The actual effect, however, would be a very small gain indeed and, practically speaking, insignificant.

There is one other point in regard to pressure which deserves emphasis. In many old-fashioned live-steam heating systems a high

pressure is used where a lower pressure would do the work. In such old systems the traps are likely to be poor and badly arranged, or there may be a conspicuous absence of traps at many points. In these antiquated plants there is much "blowing through" of the steam. It is obvious, therefore, that if the initial pressure be made as low as possible the steam waste will be reduced accordingly, for the discharge of a pipe is proportional to the pressure. Therefore a crude but paying improvement under these rough conditions is the installation of a pressure-reducing valve between the boiler and the heating system. Or if no engines or pumps are operated the boilers themselves may be run at the reduced pressure if the steam mains are of sufficient capacity.

A very common trouble discovered in heating systems is ineffective circulation of the steam chargeable to badly designed drip lines from the radiators or soils. The water of condensation should be conveyed back to the boiler room by branch and main pipes of adequate capacity. These drip pipes should have a gradual slope toward the boiler plant so that the water will flow by gravity throughout. If there is depression in the line a "pocketing" will occur, drainage will be incomplete, and the capacity of the re-

turn will be reduced. The gross effect is the choking of the circulation, necessitating the use of pressure for its re-establishment, and this often results in the otherwise unnecessary use of live steam.

Sufficient reference has been made to the re-evaporation or discharge from the radiators of a part of the drips as steam to permit realization of the fact that the return lines must be designed to carry not only water but steam as well. Consequently ample capacity of these pipe lines is an essential feature to efficiency of heating operation.

This discussion on the broad principles of heating is merely by way of introduction to our investigation of this department of the factory power-plant problem. We must here remind ourselves that our "steam doctor" has obtained the individual efficiencies of the boiler and engine plants and is equipped with full data on their everyday working performance. The heating problem is therefore attacked in the full light of its relations to the steam-making and steam-consuming equipment. The specific question next in line is as follows:—

What are the heating requirements of the plant under examination? There are several methods of obtaining this information, but they may be classed under two heads:—

(A) By actual tests under working conditions, this being the more satisfactory way.

(B) By calculation from accurate data pertaining to the particular case in hand. This is the method in general use by designing engineers, since the first scheme is of course impracticable before the plant is in operation.

(A) As we are considering a working factory we shall apply the first method wherever possible. Each plant presents a special problem as to the simplest manner of making these tests. During the investigation of the boiler plant an accurate water weigher or measuring apparatus has been set up for recording the feed water. This is always kept in readiness for use until the examination of the entire plant has been completed.

If therefore we can isolate one or more boilers to supply steam exclusively for heating, we have simply to weigh the feed water to such boilers in order to determine how much steam is consumed by the radiation. Temperatures and pressures are of course carefully observed during the heating test. This works out beautifully for the measurement of live-steam heating. Now by modification the same system can be made to record the amount of *exhaust* steam supplied

to those portions of the radiation. I have done this in the following manner:

The exhaust steam from the engine which supplied the heating system was temporarily turned out of doors through the relief valve. Then live steam from a single boiler was supplied through a reducing valve to take the place of the exhaust steam. The reducing valve was adjusted to give the same pressure in the heating system that was regularly maintained with the exhaust steam, thus duplicating the heating conditions. The water evaporated in the test boiler was recorded by the automatic weigher and thus the exhaust or low-pressure heating requirements were accurately determined.

Now since the steam consumption, and therefore the exhaust steam per horse-power hour of the engine, had already been determined, it was only necessary to take indicator cards during the heating test. From the engine horse-power hours the exhaust steam produced was readily computed. From the efficiency of the engine correct allowance (about 6 per cent in this case) was made to give the true heating value to the exhaust. It was now known exactly how much heating was required, how much was available in the form of exhaust from the engine, and consequently how much was wasted through the

relief valve in everyday practice of that season of the year.

These tests were made to discover whether it would pay to install a more efficient type of steam engine. It was found, however, that since over 90 per cent of the exhaust steam was utilized, an improved type of engine would not pay for the interest on the money required for its purchase. This result necessarily included a consideration of the fuel cost of steam in the boiler plant. If the latter had been sufficiently high a better engine would have paid for itself.

This shows how absolutely all phases of the power-plant problem are bound up together, and all must receive due consideration unless common guesswork is to be relied upon.

Another method of making an actual determination of heating demand is to catch and weigh the hot drips from the radiation. If the object is to learn the *total* amount of steam consumption it is necessary to *condense* the steam which accompanies the actual water of condensation, in order to obtain the full weight of steam fed to the heating system.

One way to accomplish this is to allow the drips to enter a coil of pipe submerged in cold water, thus forming a condenser, and then to weigh the resulting water at the cold

end. During the test (which should extend preferably over a complete run of the plant) full observations should be taken of the temperatures of the air in the work rooms, the dry rooms, etc., as well as out of doors; a recording gauge should be used or periodical readings of steam pressure taken, both at the boiler and of the steam entering the heating system; and systematic readings of the condensation should be taken and regularly recorded.

Where the flow of steam to a heating system is nearly constant and is without pulsations (as in the case of engine exhaust) the heating steam may be measured by attaching to the supply pipe a special form of steam meter. Some of these depend upon the principle of the Pitot tube and others employ the Venturi meter system. So much for actual tests under working conditions.

(B) Now supposing, under the circumstances that often occur, it is impracticable to make such trials as we have just described; we have yet a means to arrive at an approximation of the heating demand by process of computation from data.

Such method may for instance be necessary in the winter time to determine the summer heating load, or *vice versa,* and as previously stated this system is used principally by ar-

chitects and heating engineers in the design of new buildings. The amount of heat that is required to keep a room or building warmed to a given temperature depends upon two factors:

1. The loss of heat carried away by the warm air constantly escaping to make room for the fresh air admitted for ventilating purposes. With this is included the accidental escape of warm air through leaks around doors and windows, and its diffusion through walls.

2. The loss of heat by radiation through walls and closed doors and windows.

The amount of heat required to overcome the first loss depends upon the rapidity of air changes allowed in the room and upon the temperatures of the air in the room and the air outside.

Authorities compute that each adult person in a room requires at least 30 cubic feet of fresh air per minute to maintain a fair standard of purity. Hence the air changes required per hour primarily depend upon the number of workmen in the shop, together with its cubical contents. If the rooms are large and the workmen few, the necessary air changes per hour are lessened, so that this figure depends for its determination

upon local conditions.[1] Mr. J. Byers Holbrook allows "one change of air per hour for the average type of city building", increasing this allowance for corridors and first floors. Other engineers designate variously from a fraction of one change to as high as three changes per hour for different sets of conditions.

In addition to ordinary ventilation special problems arise, such as the supply of air for process work in dry rooms, etc. In such cases decisions based upon experimental data are the most reliable.

Having once decided upon the rate of ventilation it becomes a simple matter to compute the heat or steam required, and consequently also the amount of radiating sur-

[1] Example:—A room contains 1,000 cubic feet of air. The minimum outside temperature is 0 degree F. Temperature to be maintained in the room is 70 degrees. Air changes per hour allowed = 2.

1 cubic foot of air at 70 degrees weighs about 0.0745 pounds.

Specific heat of air is 0.2375.

Hence 1 B.t.u. will raise the temperature of 56.5 cubic feet of air 1 degree F.

The problem is to raise 2,000 cubic feet of air per hour 70 degrees F.

$\frac{2{,}000 \times 70}{56.5} = 2{,}480$ B.t.u. per hour required to warm the air for ventilation.

Then the amount of steam needed at atmospheric pressure will be $\frac{2{,}480}{970.4} = 2.56$ pounds steam per hour. Figuring the "rate efficiency" of the radiators at 1.8 B.t.u. per hour per degree difference per square foot and steam at 212 degrees the temperature difference will be 212 — 70 degrees = 142 degrees, and 1.8 × 142 degrees = 255.6 B.t.u. per hour per square foot of surface.

Therefore the amount of radiation required will be $\frac{2{,}480}{255.6} = 9.7$ square feet, which will warm the air for ventilation.

face that must be provided and installed on this account.

Now in addition to the requirement of warming the ventilating air we must provide for the second item, that is, the heat needed to overcome the loss by radiation through walls and windows. Of course the heat loss through a wall depends upon its thickness and its material. Holbrook uses 0.25 B.t.u. per square foot per degree difference in temperature for the average city-building wall, and states this coefficient will vary for brick walls (from 4 inches to 18 inches in thickness) from 0.24 B.t.u. to 0.68 B.t.u. The heat transferred by glass in windows also varies with conditions, but may be taken at an average value of 1 B.t.u. per square foot of surface per hour per degree difference in temperature between the inside and outside air.

These wall and window radiation losses are computed and added to the ventilation losses to produce the total heat and the consequent radiation required.

For a rough and quick calculation in factory work where the radiation is known, we may allow one-third of a pound of steam (low-pressure) consumed per hour for each square foot of direct radiating surface.

Chapter VIII

THE HEATING SYSTEM (*Continued*)

THUS far we have considered only direct radiation, that is to say, cases where the radiator coils are surrounded by comparatively still air. Now where the fan blast or indirect method is employed, a square foot of surface will give out from about three to six times the amount of heat transmitted by the same area in an ordinary radiator, depending upon the velocity of the air and the temperature differences. This system subjects itself to more accurate calculation since it is usually more or less centralized and the volume of air supplied can be directly measured.

Thus it is possible by actual testing and by computing from accurately gathered data, to obtain the heating requirements of any factory plant.

Both the engine loads and the heating loads must be obtained not only for summer and

winter but for day and night as well, and the production of exhaust steam by pumps and auxiliaries must be taken into our calculations.

It is evident that these steam and heating conditions are continually varying, and the highest economy is possible only by balancing the production of exhaust steam to our heating requirements for all conditions of operation the year round. This introduces, next to the boiler plant, the most serious and the most important part of our factory fuel problem. For a guide in its solution we may state the following working principles:

1. Substitute exhaust for live-steam heating wherever practicable.

2. Design or modify the plant so as to produce sufficient exhaust steam (and no more) to take care of the heating requirements at all times.

3. If a surplus of exhaust steam beyond the heating demand is found, it can be reduced:

a. By more economical type of steam engines and pumps running non-condensing.

b. By operating one or more of the engines condensing during such time as their exhaust steam cannot be utilized.

c. To care for the lighter heating load in warm weather, the exhaust production may

be reduced by running at a lighter load one of the non-condensing steam engines, or shutting it down entirely and shifting its work on to a main engine of high efficiency running condensing. This is a simple matter in an electrically operated plant with a main engine of sufficient capacity.

4. Where possible, always make the boiler steam do work by putting it through an engine *before* applying it to the heating system. Thus the power becomes a by-product of the fuel needed for heating.

5. Where condensing is out of the question, the amount of surplus exhaust may be reduced by making a part of the power with oil, natural gas, or producer-gas engines. Or where sufficiently cheap electric power is available, it may be drawn upon up to the point at which exhaust steam balances heating load.

6. A compound condensing engine or bleeder type of turbine may sometimes best meet the combined power and heating requirements by drawing from the intermediate stage or bleeder the exact amount of exhaust steam wanted at all seasons. Such steam has performed work in the high-pressure cylinder or stages before its application to the heating system and therefore this practice is highly economical. That portion

of the exhaust or low-pressure steam not used for heating proceeds through the low-pressure cylinder or stages and delivers additional horse-power hours before its rejection to the condenser.

This effect of conserving the heat in the steam for a maximum output of power and heating combined may also be accomplished by operating a hot-water heating system in conjunction with a condensing turbine or engine. The water is heated in passing through a special heater placed between the prime mover and its condenser, the vacuum on which is reduced as more heat is required for the radiation, and *vice versa.* Thus in very warm weather when no exhaust is required for heating, a high vacuum is carried with a consequent economy in steam consumption. In cold weather the vacuum on the condenser is reduced and more of the heat of the exhaust steam enters the water which is circulated through the radiation for warming the buildings. This plan is known as the vacuo hot-water system and will be further discussed.

7. *A non-condensing steam engine with its exhaust efficiently and wholly utilized,* when properly credited with the live boiler steam that would otherwise be required for the heating load, *will furnish a horse-power hour*

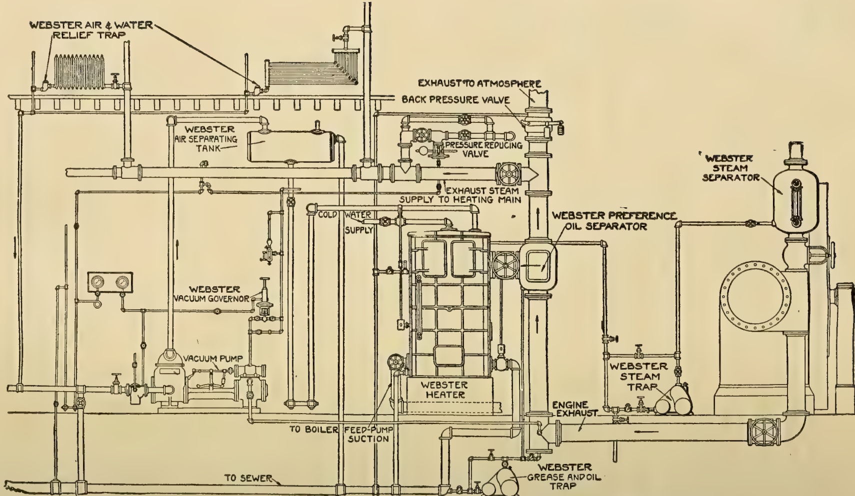

Fig. 11. Typical Layout of a Vacuum Return-line System.
(Courtesy of Warren, Webster & Co.)

for about 3/8 pounds of coal, whereas a large and refined central power-station can only approximate a horse-power hour for one pound of coal.

8. In the use of exhaust steam, the heating of the boiler-feed water should be given first preference.

Approximately one-seventh (1/7) of the steam from a boiler after it has passed through a simple engine will heat all its feed water from 60 degrees to 212 degrees. The balance of the steam (6/7) is available for other heating purposes.

9. Conserve first the exhaust steam. Then look after the return of the hot drips from the heating system. Effective oil separation is essential with the latter, in order to prevent burning of the boilers. The return of hot drips reduces the amount of scale that is deposited in the boilers.

10. Low-pressure is to be preferred above high-pressure steam for reasons of economy previously mentioned.

With this list of generalities in mind, the engineer selects a method of heating which will best solve the problems of any particular situation.

The final plans usually resolve into an adaptation of one or more recognized "sys-

tems'' which may best meet the conditions in hand.

A brief description of a few of these will be *apropos* of our previous discussion.

First, we may properly mention what is commonly known as the vacuum system, or more accurately the return-line vacuum system of steam heating. A typical installation of this kind is illustrated in Fig. 11. Its principal features comprise the following:

All the exhaust steam from the non-condensing engines and pumps is led to a feed-water heater (of the open type in this case) after passing through a separator to remove the cylinder oil. The heater absorbs sufficient heat to raise the feed water to about 210 degrees if properly designed. The large residue of exhaust steam then passes into a heating main from which pipes are branched off to the radiators and heating coils.

The condensation from these is automatically released with the accumulation of air into the drip returns by the action of special ''vacuum valves'', one of which is attached to each individual radiator or heater. The condensation and air pass downward through the return mains to a vacuum pump which maintains a pressure below atmosphere in the return lines. They are then discharged into an overhead tank, which is vented for

the separation of the entrained air, so that the condensation alone flows back into the feed-water heater to mingle with whatever make-up water may be required to supply the boilers. The inflow of the make-up water is automatically regulated to suit the varying requirements of operation by means of a tank float-valve.

If there is an over-abundance of exhaust for the supply of the radiation its escape is provided for by the back-pressure relief valve which is always a necessary part of the equipment. If, on the other hand, there is not enough exhaust to satisfy the needs of the heating system, an additional supply of live steam is automatically admitted to the heating main by means of the pressure-regulating valve shown on the drawing and so designated.

The theory of the "vacuum system", which is so called on account of the light vacuum carried in the return piping, is very largely misunderstood. Many persons think that the vacuum pump "sucks" the steam through the radiators, thus causing a positive circulation, but this is not true.

If water and air are removed from a radiator as rapidly as they accumulate and the discharge be otherwise sealed, it will act in the capacity of a condenser by the reduction

of the contained steam to water. The volume thus greatly diminished creates a partial vacuum in the radiator, causing a positive inflow of the steam from the mains, which carry a pressure approximately atmospheric. The function of the vacuum is to cause a difference in pressure between the interior of the radiator and the drip line in order to permit a rapid discharge from the former of the water and air which are continually accumulating.

Without this return-line vacuum the necessary difference of pressure would be obtained by increasing the pressure of steam *in the radiator*. This however would result in back pressure on the engine supplying the exhaust steam, thus decreasing its efficiency and capacity and adding to its steam consumption, all of which disadvantages are overcome by a well designed vacuum system.

There are a large number of successful systems of this class in the field and they differ principally in the design of the return-line vacuum valve which is attached to the radiator. These automatic valves may generally be divided into two classes, though with some exceptions.

One of these depends upon the principle of expansion and comprises the thermostatic di-

vision. This type is shown in Fig. 12. Its operation depends upon the elongation and shortening of a member or stem which closes and opens the valve. When this expansion member is surrounded by steam its greater length causes the valve to shut, preventing the escape of steam; but when water and air

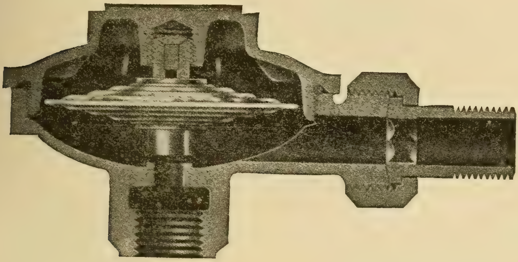

FIG. 12. DUNHAM RADIATOR TRAP, SECTIONAL VIEW. THERMOSTATIC TYPE.

collect, their slightly lower temperature causes the valve to open, thus allowing their discharge into the return line.

In the later designs of thermostatic valves, such as here illustrated, the expansion piece is of special hollow construction and contains a volatile liquid which vaporizes and forms pressure under the heating influence of the steam. By such a device the effective force of opening and closing and the stroke or mo-

tion of the valve may both be greatly magnified. It should always be carefully specified under what particular temperature conditions the valve is to operate.

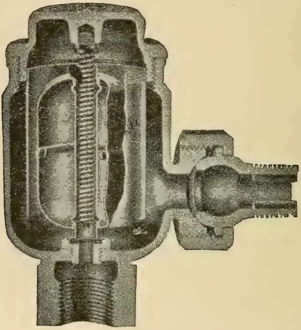

FIG. 13. WEBSTER RADIATOR TRAP. FLOAT TYPE.

The other large division of vacuum valves may be classed as belonging to the float type of steam traps (see Fig. 13). For their opening they depend upon the accumulation of water in their surrounding recess which exercises the effect of buoyancy to lift the valve from its seat. These valves are generally provided with an auxiliary port for the relief of air from the radiator. A continuous passage of air or steam takes place through this port, and when there is not enough air to satisfy the calculated leakage, uncon-

densed steam takes its place and passes unused to the return line.

A vacuum system may be applied equally well to indirect radiation or to any kind of steam-heating apparatus as a means to promote efficient circulation, and to reduce back pressure on engines and pumps when exhaust steam is used.

When both high- and low-pressure heaters are employed the high-pressure drips should not be discharged directly into the return line of the vacuum system, since the steam that will be re-evaporated in the low pressure of the partial vacuum interferes with the action of the vacuum pump to such an extent that a spray of cold water must be injected into the suction of this pump to enable it to perform its function. The secondary result is the cooling of the otherwise hot drips, sometimes to the extent of producing an oversupply of feed water with the attendant waste of heat to the sewer.

It is possible, however, to use a single low-pressure return main for both high- and low-pressure heating. This is accomplished by allowing the high-pressure trap to discharge into a receiver, the top of which is connected to a low-pressure steam supply main. Thus the "blow" or steam from the high-pressure radiator is returned for a second use, this

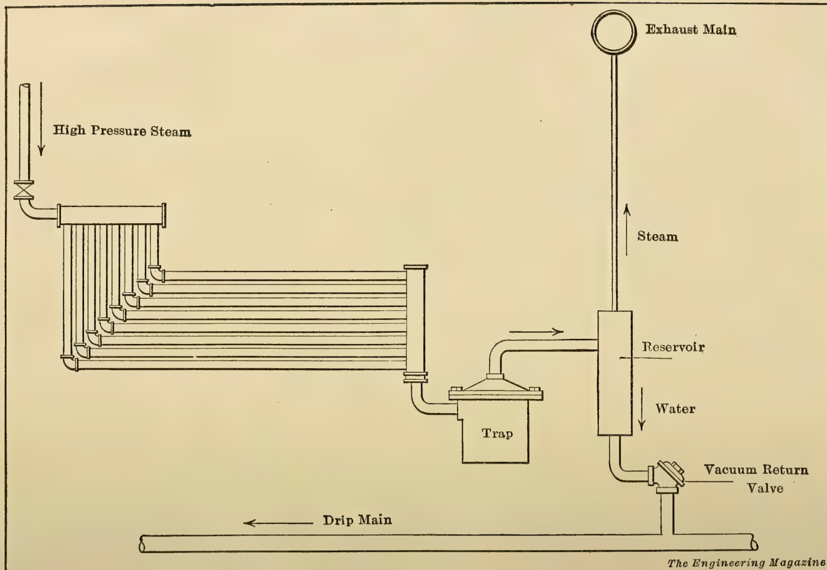

FIG. 14. SHOWING METHOD OF CONNECTING HIGH-PRESSURE DRIPS INTO VACUUM RETURN LINE.

time in the low-pressure system. The water which accumulates in the receiver is drawn off through the vacuum valve into the low-pressure return line. This scheme, which is shown in Fig. 14, not only eliminates a separate high-pressure drip main, but also results in conserving the "blow" of the high-pressure traps, which would otherwise be wasted.

Before the invention of vacuum systems the gravity system was in general use. Each radiator was equipped with a steam-supply pipe and a drip line, each controlled by a hand valve. The main drip from several heaters was often led into a common trap of the bucket or float type. This arrangement is particularly liable to air binding, and the by-pass on the trap would have to be opened on occasion to blow out the system and start it working again. Various forms of automatic air-relief valves were applied to the individual radiators, which proved helpful in general toward the relief of air binding, though they also caused much trouble through leakage and adjustment difficulties. Some, of course, were better than others.

If the trap itself became filled with air the float or bucket would not work. Then the by-pass or air cock would have to be opened

and the system "blown out" to enable it to resume operation. When the steam was shut off on a heater without closing the drip, the vacuum thus caused would tend toward "water-logging" and "hammering".

The next real step in advance was the application of the vacuum air-line system. This is the gravity system *plus* a small air pipe, attached to each radiator, and running into a main air line to which a vacuum pump or ejector is attached. The withdrawal of air from each radiator into the air line is governed by an automatic valve leading to the air line, which opens on cooling and closes when steam attempts to pass. This scheme much improved the circulation, but did not include the automatic and regulated relief of water from the heaters. This last feature was later combined with the removal of air, in the vacuum return-line system first described.

There are many variations and combinations of these several systems, but it is not our present purpose to treat of them in further detail.

The question of direct or indirect steam-heating may be decided upon the merits of any special case. The latter has the advantage of a centralized use of the steam, generally in or near the engine room, and is under

direct control of the engineer. The usual indirect system employs a blast fan which may either induce or force the air through a coil or heater whose pipes or sections are supplied with steam, although hot water may be employed as the heating medium. The heated air is then conducted in proper flues and discharged as desired in the more or less distant work rooms.

There is a tendency to supply more air with a fan-blast system than is actually needed; consequently when there is no return air-flue provided the fan may consume a wasteful amount of steam. Power also is required to drive the fan, but the additional exhaust steam thus produced may often be consumed in the coils of the fan itself, in which event only an insignificant amount of steam or fuel for power is chargeable against the fan system.

From the standpoint of operating economy (including the consideration of repairs, convenience, and a greatly reduced amount of radiation and steam piping which would otherwise have to be distributed throughout the plant) it can be readily seen that the fan-blast system has some marked advantages. The less radiating surface required is due to the velocity with which the air impinges upon it. A blast heater depending on conditions

of operation will condense from three to six times as much steam per square foot per hour as will a direct radiator in comparatively still air; that is to say, the "rate efficiency" of the blast coil is very high. This can be nicely demonstrated by directing the air blast from an ordinary electric desk fan upon the surface of a steam radiator. I have seen a cold office quickly warmed by this means.

A method of applying hot-water heating in such a manner that all the exhaust from the prime mover is utilized either for heating or producing additional power is treated in the writings of Ira H. Evans, who describes the operation of a plant so arranged under his supervision.

The turbine exhaust has two branch connections to the condenser, each under separate control. In one branch is inserted a heater through which the water for the radiating system is mechanically circulated. By changing the vacuum a greater or less amount of heat is transmitted to the heater from the exhaust steam. When no heating is required a high vacuum is maintained with a consequently reduced steam consumption for power. As more heating is needed, a lower vacuum adjustment permits the transmission of more heat to the radiation water.

Furthermore, as more steam for heating is required its quantity automatically increases with the reduction of vacuum on the turbine.

There are still other methods of heating which the engineer has at his disposal. Air or water may be warmed by passing through an economizer type of heater arranged to absorb a portion of the heat which escapes from the uptakes of boilers. Or such heaters may be separately fired, as in ordinary air and water heaters used in small work.

For the factory engineer a knowledge of all available systems and methods of heating is, of course, essential. But unless the heating design rests upon the broad principles which we have discussed, it makes but little difference which *system* is adopted as far as over-all efficiency is concerned. Primarily, minimum cost of power and heating do not depend upon a *system.* I have said *power and heating,* not heating alone. For it is impossible to isolate one from the other in an investigation directed toward the attainment of efficiency.

Success in this matter does depend first and last upon a thorough knowledge of all the working conditions in any given case, and the relations involved, which include a determination of the power and heating loads for day and night for all seasons of the year.

These data are to be had only by means of a scientific investigation such as we have here discussed. With this kind of a report in hand for basic reference, we may then mold a plan which will provide not merely for individual efficiencies of the component parts, but for a maximum economy of the whole scheme of operation all the way from the coal pile to the return drips, together with a provision for future expansion in the path of true efficiency.

Chapter IX

THE HUMAN FACTOR

OVER one-fourth of the factory's yearly coal bill is directly controlled by the fireman. In New England a single fireman will burn \$40 to \$50 worth of coal a day in a 500 horse-power plant. That is \$12,000 to \$15,000 worth of coal a year. This man receives from \$1.75 to \$2.50 per day for his work. Depending upon the skill with which he handles his shovel, he will waste or save \$3,000 to \$4,000 a year of his firm's good money.

Even with excellent physical equipment, there may prevail either high efficiency or a waste of fuel amounting to thousands of dollars per year. In my experience the range of this difference in *operation alone* extends from 48 per cent efficiency to over 70 per cent. Thus in this extreme, about 46 per cent more steam could be made from the same amount of coal, while for the same

quantity of steam 31 per cent less coal would be consumed.

Hence my introductory statement is very conservative; over one-fourth of your factory coal bill is directly controlled by your fireman.

Forty-nine managers out of fifty are quite ignorant as to the disposal of this fourth part of their coal bill. A frequent attitude on their part is: "We employ 'good men' and we can't be losing much," and "I guess you will find we are doing as well as the next fellow." In a recent case of this kind I found the firm *were* wasting over one-quarter of their fuel by *wrong operation alone.* When the matter was set right, which was quickly done, they expressed themselves as undecided whether to be pleased with the saving or to be overcome with remorse that they had been losing such a large sum of money annually for so long a time.

The "good man in charge" theory of efficiency is a sad fallacy. It cannot be depended upon without additional aid in the boiler plant—the simple reason being that no matter how "good" the man may be, unless he has a means of checking the efficiency of his plant and knows how to use it, he will be utterly in the dark as to the results he is obtaining.

In each of the other departments of a factory not only is a record kept of all material and items of expense which enter into that particular process, but a most careful account is taken of the *output* or *product* which forms the object of the expense involved.

On the other hand, most managers are content to keep a careful account of the *expense of operating the boiler plant* but have only the vaguest idea of *what this expense produces.* It is most usual to find strict records kept of the *amount of coal consumed* but absolutely no figures of any kind to show the *amount of steam this coal has produced.*

This represents the same type of transaction which takes place between school boys when they trade jack knives, "unsight and unseen". The trader knows what he is *giving* but there is always a delightful uncertainty as to what he will *receive.*

The "old-school" manager handles his boiler plant in precisely this manner. He "blows in" anywhere from $5,000 to $100,000 in good coal each year, and takes a blind chance as to whether one-quarter of this amount will produce steam or "go up the stack"! He places a "good man" in the boiler plant and keeps track of the coal consumed, but neither he nor his "good man" is able to say how much steam that coal *pro-*

duces, nor how much it *ought* to produce. To be sure they will both *guess.* But judging by the experience I have had in converting guesswork of this kind into positive information, their guess will invariably prove to be a bad one.

Managers have pretty well come to the correct conclusion that it pays to *buy* their coal carefully, and preferably on a heat unit and analysis basis, *but only a few have learned the value of measuring the product as well as the expense of the boiler plant.*

This guessing game may have been amusing in the days of cheap coal (though a bit rough on the present effort toward conservation of the resources), but in these times of high and increasing coal prices the sport is becoming expensive. It is therefore safe to predict that in the future it will be followed by only the most reckless of our captains of industry.

Some managers will resent my statement regarding their failure to measure production as well as expense in connection with their power-plant efficiency, because they keep an account of the amount of coal consumed per barrel of flour or per other unit of factory product. To these I would answer that there are so many factors other than power-plant efficiency which enter into the

fuel cost of a unit of the finished product, that a figure obtained in this manner conveys no information whatever regarding the true performance of the power plant.

Should this for a moment be doubted, allow me to cite an instance of two plants which I visited, both of which were making the same product and owned by the same company. In the one which possessed the better power equipment, the fuel consumed per unit of production was the greater amount. But this factory was handicapped by a bad layout of buildings and much more heat and mechanical energy were *demanded* for its operation. Thus the fuel per unit of finished product was in no way proportional to the efficiency of the power plant itself.

This simple case illustrates the futility attempting to accept such "over-all data" as in the slightest degree indicative of the individual performance of the power-plant department.

If efficiency is to be obtained it is necessary to bestow individual analysis upon this one small *factory* whose raw material is coal and labor and whose product is power, light and heat.

The average fireman works entirely in the dark. He only knows he must maintain the required steam pressure, and he is made to

fire as many boilers as he can, "to keep the payroll down." He has long hours—twelve hours a day most frequently, and sometimes more—and he is paid a low wage with no hope of more for better work. I know one man who fired from 16 to 18 tons of coal each day, serving a long battery of boilers.

The fire-room is usually very hot and badly ventilated and no conveniences or shower bath are provided. It is the rule to find only such appliances for safety as may be compelled by law, and these are generally inadequate.

When matters do not run smoothly our old-school superintendent is very likely to visit the fireman with added affliction in the form of violent language and dockage of the pay envelope. I have seen and heard these disheartening occurrences.

The fireman is regarded by such a man as but little better than an animal or beast of burden. *He will brag about the low wages in his fire-room while for every hundred dollars thus saved a thousand dollars in coal is being needlessly wasted,* and principally owing to this unintelligent, not to say brutal, attitude of the factory superintendent himself. Numerous examples and experiences of this kind reappear before me as I write and they form a most unattractive picture.

Combustion of fuel involves highly technical considerations. It comprises a series of more or less complicated chemical reactions which are not even yet *completely* understood by scientists who have made it their particular study, although the knowledge of the subject which is now available, and in practice, is capable of producing and does produce remarkably high efficiencies. In view of these facts, is it strange that the average factory power plant needlessly wastes a large part of the expensive coal which is annually purchased for its consumption? The principal facts to which I refer are: the complexity of the science of combustion; the low wages and long hours of the fireman; the lack of encouragement he receives from his superiors; and the ignorance and wastefulness of the manager who records the *expense* but disregards the *product* of his boiler plant.

With the exception of the fireman, I know of no other class of worker who is entrusted with the expenditure of $12,000 to $15,000 a year of the firm's money who in the first place receives so little as $2 to $3 per day, and in the second place goes wholly unchecked as to the return he makes to his company for the amount expended.

The trend is toward a widespread reform, and it is actuated by two motives. The first

motive is economy. The second motive is humanity. It would be pleasant to reverse the order of these betterment forces, and at some future time this too will be accomplished. That *economy and humanity travel hand in hand* is just beginning to be recognized, and most notably so in our own United States.

The firemen in our boiler plant will in the future be far better paid, their working hours will not be excessive, and the boiler rooms will be better ventilated. The firemen will be given a direct financial interest in eliminating the preventable waste of fuel; they will receive specialized education to enable them to produce very large economies that only await such educational incentive. They will become skilled workers who will annually save for their companies ten and twenty times the increase in their wages. They will feel the spirit of encouragement and co-operation from their superiors, and when they are taught the value of combining brain with brawn their future will become wider, brighter, and more useful.

The future for the factory *owners* will present a radically different and better view than under the present backward conditions. It will be positively known whether one-quarter of the coal is being needlessly wasted

Guesswork will become a relic of the past. Simple but reliable boiler-room accounting systems will prevail, and *there will be great wonderment as to why such a policy had not been adopted years and years before.*

It will be known in the manager's office just what evaporation the expensive coal *should* produce, and just how closely this standard of efficiency is being maintained. The fuel cost of evaporating 1,000 pounds of steam will be as much a matter of common information as the unit cost of production in the factory, or as profits and dividends at the end of the year.

Strikes and troubles with the men in the boiler plant will become very unusual, for the men will be contented. *They will share in the profits of saving coal* on a bonus or similar plan *so that they are one with the company in their interest and incentive to save money.*

Safety methods and appliances will be provided in the power plants of the future and the economic result of all these things, simple changes in themselves, will be the saving of millions of dollars' worth of coal to increase the profits of our manufacturers.

If this reform were instituted to-day, New England's manufacturers alone would add $8,000,000 to their yearly profits. This esti-

mate is conservatively based upon the annual factory coal bill in these States and upon the extent of preventable losses which I have found to exist in the boiler plants of our industrial establishments. The firemen in the boiler plants will some day share in the division as well as in the producing of these extra dividends. *The firemen can waste or save this money. It is in their hands. Therefore the sooner this is realized by owners, and the sooner they offer a just share of the proceeds to the men who work with furnace and shovel, the sooner will these great results be achieved.*

But these large and worth-while economies can be successfully gained only by means of careful and intelligent application of *system based upon scientific knowledge.* It will not be sufficient to arrange with the firemen that they shall receive, say, a quarter of the value of the coal they save for the company. A manufacturing plant never uses the same quantity of steam for two successive months or years. This constantly varies with the weather, with the output of the factory, with changes of machinery or process, with increase or decrease of the business, and with other conditions. Consequently so crude a scheme would be sure to fail.

But it *is* possible to place the boiler plant

on a productive basis in the same way in which other departments of any progressive manufacturing establishment are now handled. Measure the expense and the product, and get the unit cost of production. Weigh the coal and the water and get the evaporation per pound of coal.

It is true that there are certain variable factors influencing the standards of evaporation in different plants. But it is quite practicable to set an efficiency standard of evaporation for any given plant which shall include all of these factors, and this is already being done. Thus for any individual plant an evaporation standard may be set which shall take into accurate account the average steam pressure, feed temperature, and heating value and moisture of the coal as well as the type and design of boiler and furnace equipment.

Thus in setting the evaporation standard recently for three different mills using the same coal, for which all of the above factors were computed into the final results, I found two of the mills to require a standard of 8.8 pounds of water per pound of coal while the standard of the third was 8.3 pounds. At these standards all three plants would operate at exactly the same boiler and furnace efficiency to make the weekly bonus payable,

but a lower evaporation was required of the third because it had no feed-water economizer to assist toward the increase of the over-all efficiency. *Thus equal efforts of the firemen themselves were equally rewarded quite independently of unequal plant equipments.*

A very heavy loss of fuel today exists in a great many of our plants which is directly chargeable to running too many boilers. That is to say, the boilers are worked very much below their proper rated capacity. It is a common failure of managers, and some operating engineers as well, to run their boilers at about half their normal load. They feel it is "easier" to make steam if they have plenty of boilers on the line, but they disregard the effect on the coal bill.

Ordinarily a boiler gives the highest efficiency when operated at or above its normal rating, and one reason for this is that its grate surface is designed for the full horsepower output. Consequently when too many boilers are under steam and they are being run at say one-half their rating (a most common practice) there will be in use twice the grate area necessary for the proper air supply for the coal. With this condition, therefore, it is usual to find from 200 to 400 per cent of surplus air passing through the

grates, to the very serious detriment of economy.

One reason why this bad practice is so prevalent is that *no one knows how much horse power is being developed.* With a simple boiler-room accounting system it *will* be known just what the boilers are doing and consequently how many of the boilers *should* be steaming, and large economies may be looked for from the simple correction of this trouble.

The human factor may be strikingly illustrated in this connection. I have known a factory superintendent peremptorily to order the engineer to fire a certain number of boilers, although he had no information whatever to prompt him as to the *right* number of boilers for efficient operation. A mistake of this kind alone will in many plants easily make a difference of one-fourth of the coal bill. A little intelligent reading of a suitable boiler meter prevents the waste chargeable to such an error.

We have thus far principally considered the boilers and the firemen because, as previously indicated, the greatest preventable waste usually occurs in this department.

The chief engineer should always be in charge of, and responsible for, the entire power plant and the equipment, including

the boilers as well as the engines, generators, pumps, etc. In some plants this concentration of responsibility is not practised, and the fire-room, if managed at all, reports separately to the factory superintendent. This method or lack of method is wrong and wasteful.

If the chief engineer is properly equipped for his duties his first interest will be the efficient operation of his *boiler plant.* I must once more emphasize the important fact that the boiler plant offers a wider field for the practice of economy than the engine plant. This is universally true for any given plant where change of *equipment* is not considered.

From close contact with a great many firemen, both in and out of the boiler house, I have learned that they regard their positions merely as a step toward becoming operating engineers. They desire to get out of the boiler room just as quickly as possible. And who can blame them under the present usual order of things, which teaches them that firing is only *shoveling coal and working in a hot dirty place for low wages, and has no "future" for them?*

From these men brought up in this environment, the *chief engineers* are produced. No wonder they feel like keeping out of the fire-room when once they are advanced to

the engine room! And they do! That is, the majority do, and so the coal bill goes merrily on as big as ever. Here, then, is another human factor. The strong tendency is for the chief engineer to disregard the efficiency of his boilers because while still a fireman he was never taught the economy of skillful firing, or if by chance he did obtain a vague idea of the amount of coal he could thus save, he learned at the same time that it didn't pay to take the trouble. No records were kept which would enable the firm to know the value of a good fireman. Therefore it did not pay to be one. Most of us would feel the same way about it. This state of affairs is typical of factory power plants today.

It is easy to understand why the chief engineer does not try to produce better efficiency in the fire room. There are always of course some exceptions, but even so the work of producing efficiency is extremely difficult without such systematic boiler-room records as have already been briefly outlined.

The chief engineer may take a great pride in keeping up his engines and steam machinery. In fact he usually does. He has gone into this business because he happens to be of a mechanical turn of mind, and he likes to see smooth-running mechanisms well main-

tained. He will generally be extremely careful to have the steam valves of his engines nicely set for the best degree of cut-off and compression, and he will not put up with leaky packings or dripping pipe joints.

The explanation is again simple. *He has had to learn these things* in order to pass his engineer's license examinations. And this knowledge leads to economy in the *engine room.* In the fireman's license examination, however, the entire emphasis is placed upon *safety.* Practically no account is taken of the man's ability *to save coal.*

These things then are self-evident. The fireman must *desire* to save coal and must be *taught* how to do it; and the chief engineer must be brought to realize the importance of his personal attention to the operation of the boilers and he must have adequate means for checking their efficiency.

There is frequently an entire lack of harmony between the chief engineer and his firemen. There is in many cases a tendency on the part of the chief to hold his men "under", sometimes with the idea of preventing them from learning enough to fill his own position. This is a wrong attitude and never leads to economy. I have found inexcusable wastefulness in the fire-room under such conditions.

On the other hand, one of my friends in the operating field spends his first month in a new place making friends with his men and learning to understand them. The result is that in the plant of his present employers he has reduced the former coal consumption by about $40,000 per year. In the plant previous to this one he effected a similar economy while employing the same policy of friendship, co-operation, and the gradual education of his men.

I have already implied the necessity of *desire* on the part of operatives to produce high results, and in order to induce this desire the simplest and most direct expedient is to give them a proportional interest in the financial gains which they produce.

But in addition to this measure, a practical example of which has been quoted, the factor of *education* must not be neglected if permanent results are to be achieved.

In view of the intricacies of the science of combustion, it might readily be inferred that the education of the fireman is a difficult if not impossible task. But under proper instruction I have seen a very ordinary fireman of not over average intelligence increase his daily evaporation from 6 pounds to 9 pounds of water per pound of coal. This was accomplished in a week's time, and with the

aid of a very simple coal-and-water-recording system. Many persons successfully operate automobiles. And yet only an extremely small percentage of these would be able to explain or even to discuss the mathematics or thermodynamics of the internal-combustion engine. It is the same way with the average fireman. It is easily possible and practicable to teach him to obtain and maintain a good efficiency with his boilers and furnaces without attempting to convert him into a scientific specialist.

So far has the economy of efficient combustion come to be recognized in very recent years that I have heard it stated by eminent authorities that the fireman is becoming an institution of the past, and in the future it is well within the range of possibility that graduates of technical schools will operate the boilers of our large central stations. The work of firing in these great power stations is for the most part by mechanical means, and the regulation of the chemical process of combustion is closely guarded by the aid of scientific instruments, including sensitive draft and pressure gauges, pyrometers, and combustion recorders. It may therefore be readily comprehended in such circumstances, and especially where so tremendous a consumption of high-priced coal is at stake, that

the man who understands combustion from a scientific standpoint may easily save his high salary every few days by the percentage of economy which his knowledge will enable him to effect, even if that *percentage* is very small.

In these great centres of concentrated power production the mechanical equipment and design approach perfection to within a reasonable degree. But these constitute only one factor of the *two* which must ever be united for success in the development of efficiency.

The other factor is *operation.* And this is purely a *human* factor, which unless so regarded will render unavailing all the science and expense which has entered into the design and installation of our elaborate modern power plant.

This point, which is only beginning to receive its due recognition, cannot be over-emphasized. *Efficiency equals suitable equipment multiplied by operation.* And since operation consists essentially and exclusively of the human element, we may set it down that *Efficiency equals the product of Equipment multiplied by the Human Factor.* This may be a new formula for efficiency, but it is perfectly self-evident and it has been demonstrated in practice thousands of times un-

failingly. Personally I have never investigated a factory power plant where I have not found that Efficiency = E x H, in which E equals Equipment and H equals the Human Factor.

If the reader desires a further proof let him test the formula by process of elimination, and before he finds a single case that does not perfectly conform he will be so exhausted with his labors that he will be ready to admit its value.

It is natural that in the *past* when equipment was crude our attention should have been almost solely concentrated upon *invention and design of power apparatus.* But mind as well as matter suffers from the effect of inertia. So that at the *present* time, when proportionately more can be accomplished by working with the *human factor,* our minds by virtue of inertia alone are carried along the old track of invention and design and we are inclined to neglect the *now more important field.*

Doubtless we shall still further improve our power machinery, but the margin for this kind of improvement is now limited, and the time is past due when the pendulum should begin to swing in the direction of the other element of efficiency, that is, toward the *human factor.*

We have briefly discussed the personnel of the operatives, but the human factor as related to the *management,* including the president and board of directors, counts *more* for or against efficiency *than all the other elements combined.* They are the brains of the industry, and if they fulfill their proper function, they will directly control its policies in every important phase of its activity. *The personal ideals of the president are reflected in, and permeate, every department of his factory.* This applies to both large and small industries and is a matter not alone of theory, but of direct and close observation. Over and over again, without fail, have I seen this truth demonstrated.

If the power plant is allowed to drift along in a wasteful, haphazard and often dangerous manner this state of things is directly chargeable to the management "higher up." *They alone are responsible for the operation of their plant and it is entirely within their control to alter these conditions.* If the president and his board consider the power, light and heating as unimportant details, these matters will likewise be so considered by the factory manager, the master mechanic, the chief engineer and the firemen; and this atmosphere of carelessness, often unconscious-

ly created, will invariably lead to a condition of wastefulness and inefficiency in the power plant.

In defence of the management it may be said that there do at times prevail such a set of circumstances that power-plant economy must be sacrificed in favor of other and more important considerations.

Such a period of pressure is never a lasting one, but if the trend is in this direction, it is advisable to make a determined effort to attack the power problem in the immediate future, otherwise operation and equipment will so rapidly deteriorate that only a disproportionately large expenditure of time and money will be able to re-establish normal efficiency standards.

I cannot conscientiously close this chapter without at least calling the attention of my readers to the importance of safety in the power plant. This naturally is an element of the human factor, and to it distinctly applies the great truth previously set down: that *"humanity and economy travel hand in hand", and where the one exists the other will be found.*

So much has recently been said and written in connection with the rapidly increasing "Safety First" movement that I shall not attempt to duplicate any of these efforts ex-

cept in so far as they directly apply to the factory power plant. But I do earnestly desire to call particular attention to the following matters which deserve the thoughtful examination of all power-plant owners and managers.

In case of the blowing out of a boiler tube or similar rupture, the steam from all the other boilers on the same main will rush back into the injured boiler and the resulting damage will be vastly magnified, with the usual or old-fashioned method of piping. This action may be prevented by the use of a non-return stop valve inserted on the lead between each boiler and the steam main. This will automatically close and shut off the ruptured boiler. Every plant consisting of more than one boiler should be equipped with this valve, of which a number of reliable makes are now available. These valves should be used in shutting down and "cutting in" boilers to insure their constant working condition.

A frequent cause of boiler explosions is the "cutting in" of a boiler to the steam main before its pressure has reached that of the other boilers. The non-return valve renders this occurrence an impossibility. Consequently these appliances cannot be too strongly recommended.

There have been cases where a man who was cleaning or repairing the inside of a dead boiler was literally cooked to death by the entrance of steam from other boilers. This may be caused by the failure of the steam valve if there is but one on the lead between the boiler and header. Hence *two* valves are recommended, principally as a good form of life insurance for the men who have to enter the boilers. And this should be sufficient reason for the recommendation although there are other practical operating advantages.

The two-valve idea applies with almost equal force to the blow-off piping, and for the same reason, although in this case owing to leakage and other troubles, two valves between each boiler and the blow-off main *are* usually provided.

Sometimes the unfortunate man inside of the boiler suffers his appalling fate because of the carelessness of another man outside who, unaware of his companion's location, opens a steam or blow-off valve connecting to this boiler. The best preventive for such accidents consists in the use of a locking device for both the steam and blow-off valves. This contrivance is simply a special form of wire basket which in use envelops the valve wheel. It is locked in position and

the man who is to enter the boiler takes the key with him, thus insuring his own safety. These contrivances are recommended by the American Museum of Safety and are very inexpensive. It is hardly necessary to say that their use is recommended except on a steam lead which is provided with a non-return valve.

Occasionally an engine, through water in the cylinder, may blow out its cylinder head, or a main steam line may burst. Either occurrence results in a violent strain in the boilers owing to the sudden drop in pressure and the consequent development of steam with abnormal rapidity. Other more complicated reactions enter, and the ultimate result may be a boiler explosion. The prevention of this class of accident may be by the employment of safety stop valves; or by the installation of a main stop valve, hand operable from a safe location. The former is the more thorough method and accomplishes its purpose by automatically shutting off the boilers from the steam main when the sudden flow of steam causes this abnormal condition. The only objection to this method is that these special valves, which combine the two functions of stopping a sudden flow of steam either out of or into a boiler, are more or less complicated and must

be used daily and carefully attended to to insure a continuously fit condition.

The second method is simpler though it does not cover the safety of that portion of the steam header which lies between this device and the boilers. All that is needed, however, is a good substantial gate valve whose stem is connected by chain or gearing in such a manner as to afford quick closing from the floor of the boiler room. This plan is illustrated in Figs. 15, 16. When an accident occurs of the class indicated, the engine-room staff are quite free to make their escape in the easiest manner, while a fireman on the *safe side* of the dividing wall closes the main stop valve without the slightest risk to his own person.

Other combinations of these two general methods of protection are also possible and they may best be chosen to suit the existing local conditions.

Safety Fire Doors

One of the most common of accidents is the burning and blowing out of boiler tubes. This may result in loss of life, but a serious scalding at the least may be expected. The safeguarding of such damage is accomplished by the use of inward opening or safety-latch fire and cleaning doors. Since they with-

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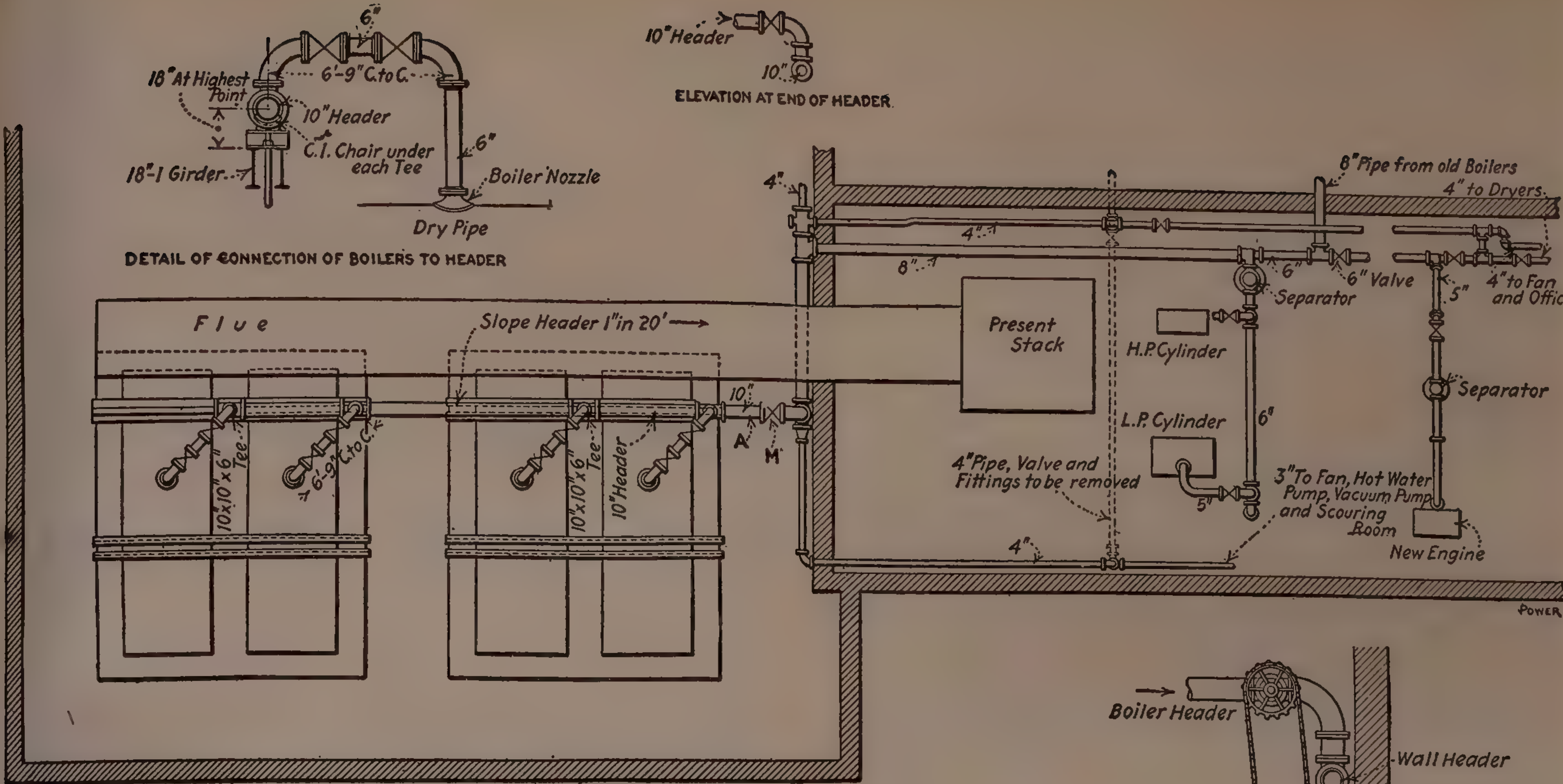


FIG. 15. PLAN OF SAFE STEAM PIPING IN A BOILER PLANT

Designed by the author in accordance with the second method described on Page 224, Chapter IX. Each lead is fitted with a non-return valve in addition to a stop gate valve. The steam main has a stop valve hand-operable from the floor on the safe side of the wall in case of accident in the engine room. Reproduced by the courtesy of *Power*.

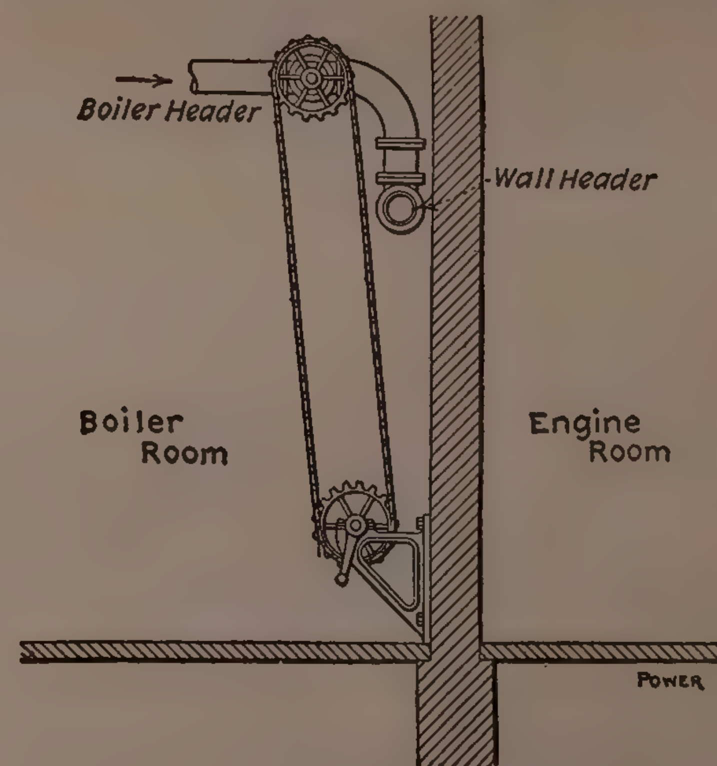


FIG. 16. DEVICE FOR OPERATING MAIN STEAM VALVE

stand the pressure of the escaping steam it is entirely confined within the boiler setting, and therefore finds for itself a natural and harmless vent through the smoke connections.

In many boiler rooms a source of danger lies in the very confined space or alley allowed at the rear of the settings. In case of a bursting blow-off pipe or fitting it is difficult for a man in this place to make a safe escape. The remedy is largely one of initial design. Ample passages for free exits from every part of a boiler room should be provided. An alley at least four and, better yet, six feet wide should be planned at the rear of the boilers, and similar spacing should be made between the various batteries.

The back combustion arch is a point of danger almost universally disregarded. The heat at this location renders the arch a normally weak structure. Should a man, in passing over, break through he would be burned alive. Yet when I advised an engineer in charge to provide against this danger by a small iron grating his answer was that "Nobody but a fool would walk over a rear combustion arch." Perhaps he was right, but if so I too am included in his list of one of the numerous classes of fools.

Those arches belonging to the type advo-

cated by the Hartford Steam Boiler Insurance & Inspection Company are the safest and strongest. The brickwork is reinforced by cast-iron beams which are themselves protected from the heat. In any case it is a simple and inexpensive matter to prevent an awful possibility by laying a small iron grating bridging over the rear combustion arch.

For the quick and safe operation of valves, there should be plain marks of some kind to indicate the nature of the piping. Some engineers resort to the simple use of markers or signs attached to the principal valves. Others adopt the more elaborate method of coloring the piping, employing a separate color for each individual class of service. One or the other of these systems should be installed in every plant wherein the management places a proper value on safety and efficiency. In many cases it is necessary to get quickly to the top of the boilers when delay may result in damage or accident. A great many plants have no provision for such contingencies. A permanent iron ladder should be secured in position for quick and easy access to each battery of boilers.

In the engine room the prime requisites for safety are: good light and ventilation, careful guarding of fly-wheels, belts, pits and

other points of danger, and the application of safety engine stops.

The last alone should require further discussion and that may be brief. A good installation prevents the engine from running away in case of accident to the governor or gearing. The bursting of a fly-wheel is often as serious in its results as a boiler explosion with its attendant loss of life and property. The safety stop is designed to prevent such an occurrence.

Furthermore, should a worker become entangled in the machinery in any part of the factory, the engine or engines can be instantly shut down by pressing a button close by the location of the accident.

There are many further details that might be mentioned, but it is desired to set forth only those special features making for safety which though most essential are too frequently neglected.

We may in the near future expect the enactment of better boiler laws in New York State which shall govern both operation and design. These laws will compel the adoption of practically all the recommendations which I have just made. An effort is also being made for the adoption of Federal laws which shall provide equal protection to boiler users in every State of the Union. This is a most

necessary measure. In some States the legislation is so absolutely inadequate to the effectual enforcement of safety in operation that hundreds of boiler plants are as dangerous as hidden bombs.

The lap-riveted boiler is essentially of weak design and the nature of its construction is such as to cause a constant process of weakening along the longitudinal seams. This is thoroughly understood and admitted by engineers, insurance companies, and designers; and yet many such boilers are in use under crowded sidewalks in New York City today. *All legislation for improved protection should cover the use of present boilers, that is to say, of old installations as well as new. Otherwise the present dangers will continue for years to come in spite of the new laws.*

The American Society of Mechanical Engineers through a competent committee have prepared a preliminary report and recommendations for sound legislation for better boiler protection, and this should receive the loyal support of all users, makers and engineers.

For the conclusion of this chapter on the Human Factor I may advisedly repeat that which I have already set down.

Efficiency is ever the product of equipment and operation.

As for *equipment,* while we may still look forward to improvement, the gain in this direction is bound to be slow, for the margin of betterment is now limited in view of the best that is already attainable. *But in the direction of operation, the outlook offers a great future. Great for the factory owners and managers and great for the men who work in their power plants. Operation deals exclusively with the human element, and the degree of efficiency betterment which we shall obtain will depend upon the intelligence and upon the amount of love for humanity which we shall weave into our efforts and express in practical achievement.*

The pendulum has already begun to swing in this direction, and as it acquires acceleration we shall see more and more clearly the truth, which in my own limited and special field of work has been so forcibly brought home to me.

For the operation of this law (for such I believe it to be), in the world-wide field of industry, I would sincerely recommend the reading of a very remarkable book which has proved an inspiration to so many besides myself. It is "The New Industrial Day", written by William C. Redfield, now Secre-

tary of Commerce and Labor in the United States cabinet.

To return to my lesser and more contracted subject of Factory Power Plants, I would call to your final attention the fact that the man who works in your fire-room can save or lose one-quarter of your yearly coal bill. Your chief engineer holds additional gains or losses in his hands. And—well, what are you doing about it?

Chapter X

EFFICIENCY SYSTEMS FOR BOILER PLANTS

IF high results are to be obtained, an efficiency system must be designed according to the human equipment of a plant.

Thus in a factory where the superintendent is a mechanical engineer, or where the chief engineer is a well-educated man, a finer and more comprehensive system may be adopted than would be practicable in the case of "ordinary" factory conditions, where no one is employed who is competent to calculate even the simplest kind of power-plant problems. In the latter type of plant a system must be devised which will reduce the necessity for calculations to the absolute minimum.

With this object in view I have standardized the efficiencies of three plants operated by one company in such a manner that the only calculation required at the end of each

day is the simple division of one number by another. In other words, it is only necessary to obtain the actual weight of coal consumed and of the water evaporated in order to know the efficiency of operation and whether or not the bonus has been earned.

Since the evaporation per pound of coal by itself signifies nothing whatever as to the efficiency obtained, it is necessary to compute into the "standards of evaporation" all the affecting factors as they are found by careful investigation to exist in any given plant where such standards are desired.

These other factors are: the average heat value per pound of dry coal, the average percentage of moisture in the coal as delivered to the fire-room, the average temperature of the feed water, and the average steam pressure maintained. Of course these factors will vary slightly in the same plant from time to time, but for commercial purposes they will not vary sufficiently to alter the value of the system. If some change of coal is made or if a new heater for instance changes the temperature of the feed water, the standards given may be modified to meet these new conditions by a simple percentage factor, as will be noted.

The computation and notes on the use of the evaporation standards for the three fac-

tory plants above mentioned are given below, together with the bonus system laid out for this case.

COAL STANDARD

The coal used at this plant contains an average of 14,300 B.t.u. per dry lb. (Coal in test contained 14,392 B.t.u. and 6.33 per cent moisture.)

Moisture assumed to average 6 per cent.

Then the net heating value of the average coal will be obtained as follows:

Total heat in a pound by deducting the moisture—94 per cent (14,300 B.t.u.)	13,442 B.t.u.
Heat consumed by evaporation of 0.06 lb. moisture and raising same from 72 degrees to a flue temperature of 452 degrees = 0.06 [(212—72) + 970 + 0.48 (452 — 212)] = ...	74 B.t.u.
Net available B.t.u. per lb. of coal weighed in fire-room..............	13,368 B.t.u.

EFFICIENCY STANDARD

With your equipment a fair standard of efficiency with proper handling of fires would lie between 65 and 71 per cent. This lower mark is readily obtainable without great effort, and its maintenance will result in a large saving over the present mode of operation. In accepting this standard it is to be noted that when the boiler test was made, No. 1 mill came within one point of the standard, although the day before I found the same boilers being operated at about 50 per cent efficiency. Raising the efficiency from 50 per cent to 65 per cent means a saving of $\frac{(65 - 50)}{(65)} = 23$ per cent of fuel.

I therefore recommend 65 per cent as the standard of efficiency at all three mills, although I believe this mark will be passed after practice with the accounting system.

I have converted this standard into a figure of *actual evaporation per pound of coal made separately for each plant.* This will eliminate all computation with the exception of dividing the water by the coal at the end of each day.

COMPUTATION OF EVAPORATION STANDARDS

Mill No. 1

The standard for this plant is based on the feed water from the economizer entering the boilers at an average of 230 degrees.

A change of 10 to 11 degrees in the entering feed water affects the standard 1 per cent. Thus if the feed water, by improving the arrangement of the feed-water heaters at this plant, should become 250 degrees instead of 230 degrees, it would be right to raise the standard of evaporation by 2 per cent.

However, with 230 degrees and 100 lb. boiler pressure, the factor of evaporation for this plant is 1.02.

The efficiency standard of 65 per cent therefore calls for an actual evaporation of *8.78* lb. of water per lb. of coal as weighed in the fire-room, deduced as follows:

$$0.65 = \frac{\text{Actual evaporation} \times 1.02 \times 970}{13{,}368}$$

$$0.65 = \text{Actual evaporation} \times 0.074$$

$$\frac{0.65}{0.074} = \text{Actual evaporation} = 8.78$$

Hence we have the *standard of evaporation for No. 1 mill, 8.8 lb.*

This is found simply by dividing the actual water by the actual coal consumed.

All the other factors have been taken care of in the setting of this standard.

For your convenience, however, should a change occur in the average temperature of your feed water, simply add or subtract one per cent to the evaporation standard for each 10 degrees change in the feed water.

Mill No. 2

The evaporation standard for this plant is based on 100 lb. steam pressure and 179 degrees feed-water temperature. Its calculation and value are given below.

$$\frac{\text{Actual evaporation} \times 1.0737 \times 970}{13{,}368} = 0.65$$

$$\text{Actual evaporation} \times 0.078 = 0.65$$

$$\frac{0.65}{0.078} = \text{Actual evaporation} = \mathit{8.33\ lb.}$$

Mill No. 3

The standard for this plant is based on the same steam pressure and feed-water temperature as at plant No. 1. That is, 100 lb. pressure and 230 degrees feed-water temperature.

The actual evaporation standard is therefore 8.8 lb.

When the feed thermometers recommended are attached the temperature of the water entering the boilers can be ascertained; and, as before directed, a change in the evaporation standard of one per cent for each 10 degrees difference in the feed water should be made. That is, the evaporation standard should be increased one per cent for each 10 degrees rise in the average feed temperature, and *vice versa.*

EFFICIENCY SYSTEM FOR THE BOILER PLANTS

Bonus

For each week during which the evaporation standard of 65 per cent is maintained I would recommend

that ——— dollars be added to the wages of the firemen of such plant.

Standards for Earning of Bonus

To earn this bonus the actual evaporation must be equal to the following:

At Mill No. 1—8.8 lb. = Actual water ÷ actual coal.
At Mill No. 2—8.33lb. = Actual water ÷ actual coal.
At Mill No. 3—8.8 lb. = Actual water ÷ actual coal.

In order to encourage still further improvement, I would further recommend that for those weeks when 70 per cent efficiency is maintained the fireman of that plant receive a bonus of ——— dollars per week. To earn this extra bonus the actual evaporation must be equal to the following

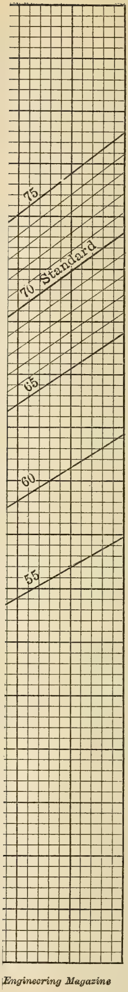
Standards for Extra Bonus

At Mill No. 1—9.47 lb. = Actual water ÷ actual coal.
At Mill No. 2—8.97 lb. = Actual water ÷ actual coal.
At Mill No. 3—9.47 lb. = Actual water ÷ actual coal.

As before stated, the three plants are averaging far below the 65 per cent standards at the present time, and the company will save a very considerable amount of coal yearly by holding to the above standards. (See other sections of this report.)

On a conservative estimate based upon your present low efficiencies and large coal consumption you will save ten times the bonuses paid to your firemen for the maintenance of the 65 per cent efficiency standards.

Consider now the case of a factory in New England where the management of the power plant was under the supervision of a mechan-



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ical superintendent, a technically trained engineer. It was possible here to install what might be termed a direct-acting efficiency system. Here the management determines for itself the variable factors entering into the computation, and then by means of a curve (illustrated in Fig. 17, inserted facing this page) determines the efficiency of the boilers and furnaces.

This system entails a little work of calculation, which, however, becomes easy as it is performed regularly at the end of each day or week, and when a slide rule is used can be done in three minutes' time. This method is more accurate than the one first described, since all variations in the feed temperature, the steam pressure, and the heat value and moisture of the coal enter immediately into the calculation of the efficiency instead of being used as a combined average factor as is the case when a "standard evaporation" is set. In other words, with the system now under discussion the man in charge must work out and apply the factor of evaporation, keep a check on the heat value and moisture in the coal received, and apply the resulting data to the efficiency curve.

With this more technical method thermal efficiency itself is the standard, and for the

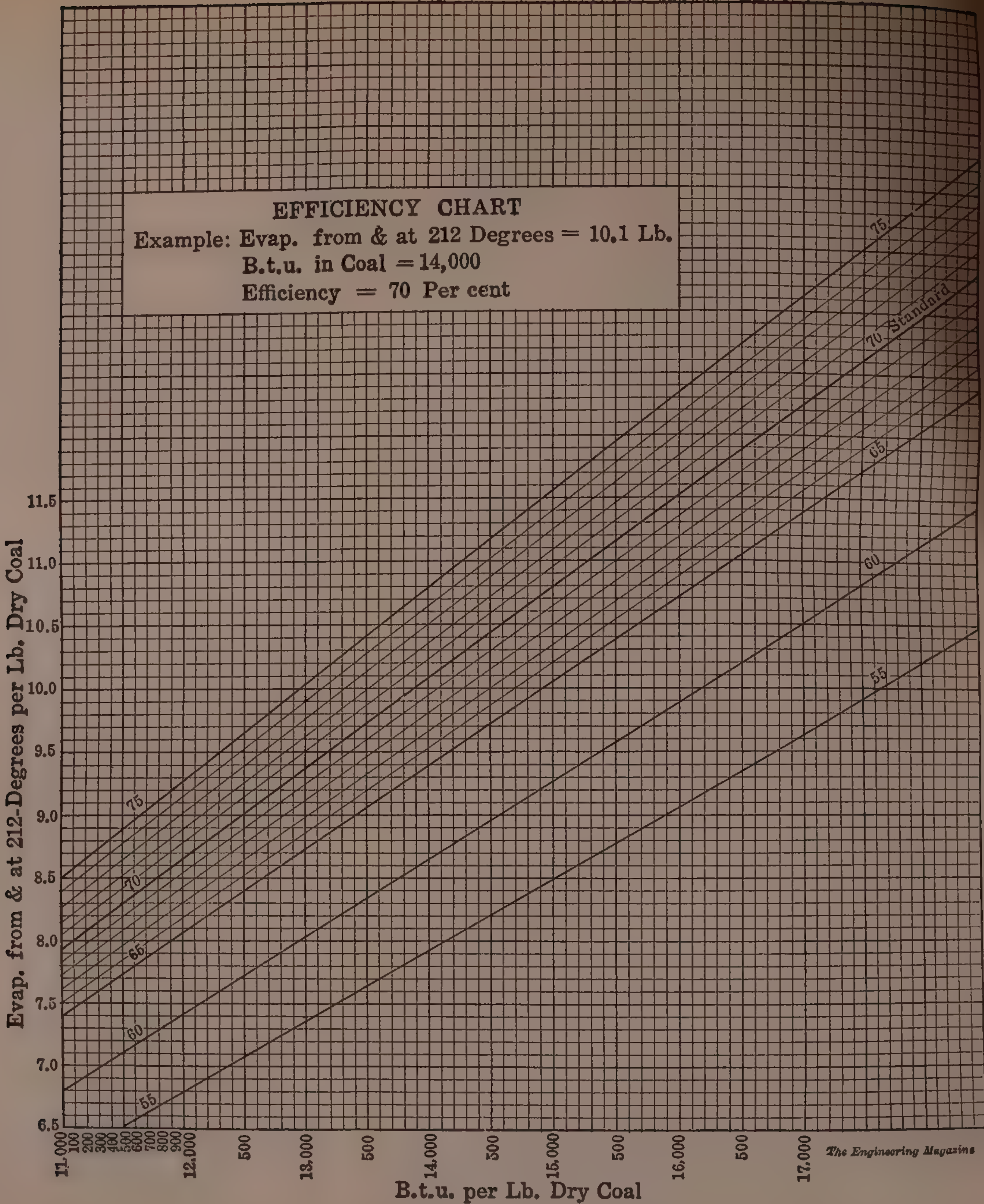

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FIG. 17. EFFICIENCY CHART

plant in hand an efficiency was set which several boiler tests proved to be readily obtainable under actual conditions of operation and equipment.

For weighing the water fed to the boilers a Venturi meter was installed and the coal is weighed as it enters the fire-room in hand cars which run over a track scale. Each car holds 1,000 to 1,500 pounds of coal.

The result of careful checking of the work of the boilers has shown an improvement of about 20 per cent in the evaporation for the same consumption of fuel.

For details of the efficiency and bonus system recommended at this plant I quote the following section of my report on this subject which may serve as a guide for the correct working out and application of such a method to any plant for which its use is suitable.

It is to be noted that in setting the *amount* of bonus the amount of saving to the firm should be directly considered, and the underlying idea should be to make the additional sums paid to the firemen a percentage of the savings produced by their increased efforts. Incidentally, these greater efforts relate far more to thoughtfulness and care than to muscular exertion. In fact, since less coal will be handled, the tendency is toward a reduction of physical work.

BOILER EFFICIENCY SYSTEM

(From Author's Report)

1. Determine the calorific power of the average run of coal by sending a sample of each of several car loads to your chemist.

With this figure in hand, and setting 70 per cent boiler and furnace efficiency as a fair standard of operation, you can use the table on a following page to determine from your coal and water records whether this efficiency is being maintained in the operation of your boiler plant.

Thus with 14,000 B.t.u. in the coal you should get an *equivalent evaporation of 10.1 lb.* per lb. of coal, etc.

2. While it is a valuable test to compare the separate efficiencies obtained by the different shifts of firemen, I would, for the sake of gaining co-operation among the men, bunch the results of the week's work and then pay a lump-sum bonus to be equally divided among the whole force. This makes each shift help the other shifts by leaving the fires in good condition, etc., and promotes good feeling.

If any one shift or man is inefficient, the eight-hour records will discover the fact and they may be trained to better work or dropped if necessary. Under this system the firemen themselves will be interested to eliminate the inefficient men, since their bonuses are thereby reduced.

Amount of Bonus

As insurance on your coal bill, it will pay to spend $260 a year to keep the efficiency up to 70 per cent.

In past tests I found the efficiency ranged from 59 to 71.4 per cent, and the best efficiency (71.4

per cent) was obtained when the boilers were run at about normal rating and not forced, and forcing is unnecessary with your ample boiler capacity.

This great difference in possible efficiencies easily warrants the expenditure of $260 a year for maintaining the 70 per cent standard.

As the boilers were run principally under forced conditions, the efficiency was about 60 per cent, corresponding to a coal bill of $14,000 a year. By increasing the efficiency to 70 per cent the coal bill will become $12,000 a year, making a saving of $2,000 annually. Hence $260 a year is a very moderate amount to insure this saving which will be produced by maintaining the efficiency standard of 70 per cent as recommended.

A further increase from 70 per cent to 71 per cent efficiency means a saving of 1-71 of the coal bill of $12,000 per year, that is, $169 per year.

From 70 to 72% saves $\frac{2}{72} \times 12{,}000$ = $333 per year.

From 70 to 73% saves $\frac{3}{73} \times 12{,}000$ = $483 per year.

From 70 to 74% saves $\frac{4}{74} \times 12{,}000$ = $644 per year.

From 70 to 75% saves $\frac{5}{75} \times 12{,}000$ = $800 per year.

By allowing one-third of these *additional* savings over those produced by maintaining the 70 per cent standard to be paid the firemen in weekly amounts, the table of Bonus Rates on the following page can be used for payment. Thus when the firemen receive an increase of $1 the firm receives $2 in coal saved. No bonus is paid under 70 per cent standard, the attainment of which brings $5 a week to the firemen with a corresponding profit to the firm. The bonus for efficiencies above 70 per cent is given in the table.

EFFICIENCY STANDARDS

B.t.u. in Coal	Efficiency Standard	Evap. per lb. of Coal from and at 212 degrees
13,000	70 per cent	9.39
13,500	70 per cent	9.75
14,000	70 per cent	10.10
14,500	70 per cent	10.46

(For other efficiencies use blue-print curve chart, Fig. 17.)

Note: This table and curve are based on dry coal. If the coal contains 2 per cent moisture, add two points to the efficiency shown; for 3 per cent, add three points, etc.

Bonus Rates

Divide among the firemen each week the following sums, depending on efficiency shown by the chart of evaporations, and efficiency.

	Bonus
Under 70 per cent	0.00
70 per cent	$5.00
71 per cent	6.08
72 per cent	7.13
73 per cent	8.10
74 per cent	9.12
75 per cent	10.12

Note: For determining efficiencies other than 70 per cent use the efficiency chart on preceding page, (see the folding insert, Fig. 17) which eliminates calculation and saves time.

If the efficiency as determined by any proper system should at any time fall below

the standard, the first thing to do is to have a calorific test and analysis made of the coal to determine whether the fault lies in the fuel. If its heat value has not become low enough to account for the reduction of the efficiency as indicated by the system, and if its composition does not show bad clinkering properties,[1] then the loss is directly chargeable to the operation of the boiler plant itself, and may be found due to bad firing, dirty tubes, scale in the boilers, or some matter directly connected with the human factor. By these means the efficiency of the boiler plant is directly controlled by the factory management.

In another plant where the steam pressure and feed temperature are practically constant quantities, it is only necessary to set a standard of actual evaporation based on good standard coal. Since the coal used in this case is anthracite buckwheat, which varies principally only according to its ash content, it is a simple and inexpensive matter to make frequent tests to determine its comparative steaming value. This is done by the company's regular chemist by burning out the combustible from a sample and weighing the resulting ash. The heat value of this

[1] Clinkering is caused by an ash of low fusing point, and is also associated with a high sulphur content in coal.

NOTE: With coal at $1.00 per ton you need an actual evaporation of only 1/2 lb. to make your cost of evaporation 9.5 cents.

coal is approximately proportional to the percentage of combustible, so that an excellent check on the calorific power of the coal is easily and cheaply obtained. In this way a good record can be kept to show the class of coal being purchased as it is received.

By the employment of this simple checking system, together with daily evaporation records from the boiler room, this plant has reduced its fuel cost of steam by nearly 20 per cent.

In still another plant where a number of different grades of anthracite coals were available at widely varying prices, and where conditions made it advisable to mix a certain percentage of soft coal with the hard, I applied the following system after equipping the plant to handle these fuels.

The object of the system was to place the determination and control of the commercial efficiency in the hands of the client, and to do so in a direct and simple manner which would entail a minimum of calculation, and at the same time give scope for a wide range of experimenting with different coal mixtures and prices, with always a definite knowledge of the result of each trial.

The coal was weighed in equal loads, a standard net weight being selected to eliminate the necessity of adding up different fig-

TABLE I—COAL MIXTURE TABLE

Cost per Ton of Various Mixtures

One (1) Part of $3.50 Coal Mixed with:

8 Parts of	6 Parts of	4 Parts of
$2.25 coal = $2.39	$2.25 coal = $2.43	$2.25 coal = $2.50
$1.40 coal = $1.63	$1.40 coal = $1.70	$1.40 coal = $1.82
$0.90 coal = $1.19	$0.90 coal = $1.27	$0.90 coal = $1.42

Find the cost of your coal mixture from this table. Then from Table II note your cost of evaporation. No calculating is necessary.

TABLE II—COST OF FUEL FOR 1,000 LB. OF STEAM FROM AND AT 212 DEGREES

The Results Are in Cents and Decimals of Cents

Delivered Price of 2,240 Lb. of Coal	Actual Evaporations Per Lb. of Coal as Fired p = 40 lb. T = 190 degrees F = 1.05										
	9½	9	8½	8	7½	7	6½	6	5½	5	4½
$3.50	15.7	16.5	17.5	18.6	19.8	21.2	22.9	24.8	27.0	29.8	33.0
$3.00	13.4	14.2	15.0	16.0	17.0	18.2	19.6	21.3	23.2	25.5	28.4
$2.50	11.2	11.6	12.5	13.3	14.2	15.2	16.4	17.7	19.3	21.2	23.6
$2.25	10.1	10.6	11.3	12.0	12.8	13.7	14.7	15.9	17.4	19.1	21.2
$2.00	9.0	9.5	10.0	10.6	11.3	12.1	13.1	14.2	15.5	17.0	18.9
$1.75	7.8	8.3	8.8	9.3	9.9	10.6	11.4	12.4	13.5	14.9	16.5
$1.50	6.7	7.1	7.5	8.0	8.5	9.1	9.8	10.6	11.6	12.7	14.2
$1.25	5.6	5.9	6.3	6.6	7.1	7.6	8.2	8.9	9.7	10.6	11.8
$1.00	4.5	4.7	5.0	5.3	5.7	6.1	6.5	7.1	7.7	8.5	9.5

DIRECTIONS: Opposite price of coal you are burning and under the actual evaporation per lb. as determined by water weigher, you will find your cost for evaporating 1,000 lb. of steam from and at 212 degrees.

EXAMPLE: Suppose you have an actual evaporation of 6 lb. of water per lb. of coal at $1.50 per ton. Cost of evaporation will be 10.6 cents.

NOTE: With coal at $1.00 per ton you need an actual evaporation of only 4½ lb. to make your cost of evaporation 9.5 cents.

ures. The water was weighed in an accurate feed-water weigher and the result given in pounds. Dividing the total water by the total coal gave the actual evaporation. Applying this actual evaporation to Table II prepared for their use (with *their* average factor of evaporation computed into the results), and co-ordinating the price per ton of coal mixture burned, they could read without computation the fuel cost of evaporating 1,000 pounds of water from and at 212 degrees.

The cost of the coal mixture was also made available from Table I without calculation. Thus any amount of experimenting with coal mixtures and coals of different prices could be done and the results immediately known from the tables. By the use of this simple system this client reduced the fuel cost of steam from 15 cents down to 9 cents per 1,000 pounds from and at 212 degrees, and without the application of scientific information after the method was once put into practice.

Another simple but very useful instrument in connection with boiler efficiency work is the cost of evaporation curve. The curve illustrated in Fig. 18 was plotted by Mr. S. Milton Clark, M. E., for the *Isolated Plant.* This embodies in graphic form the essentials of Table II already given.

Evaporation Pounds Water per Pound Coal

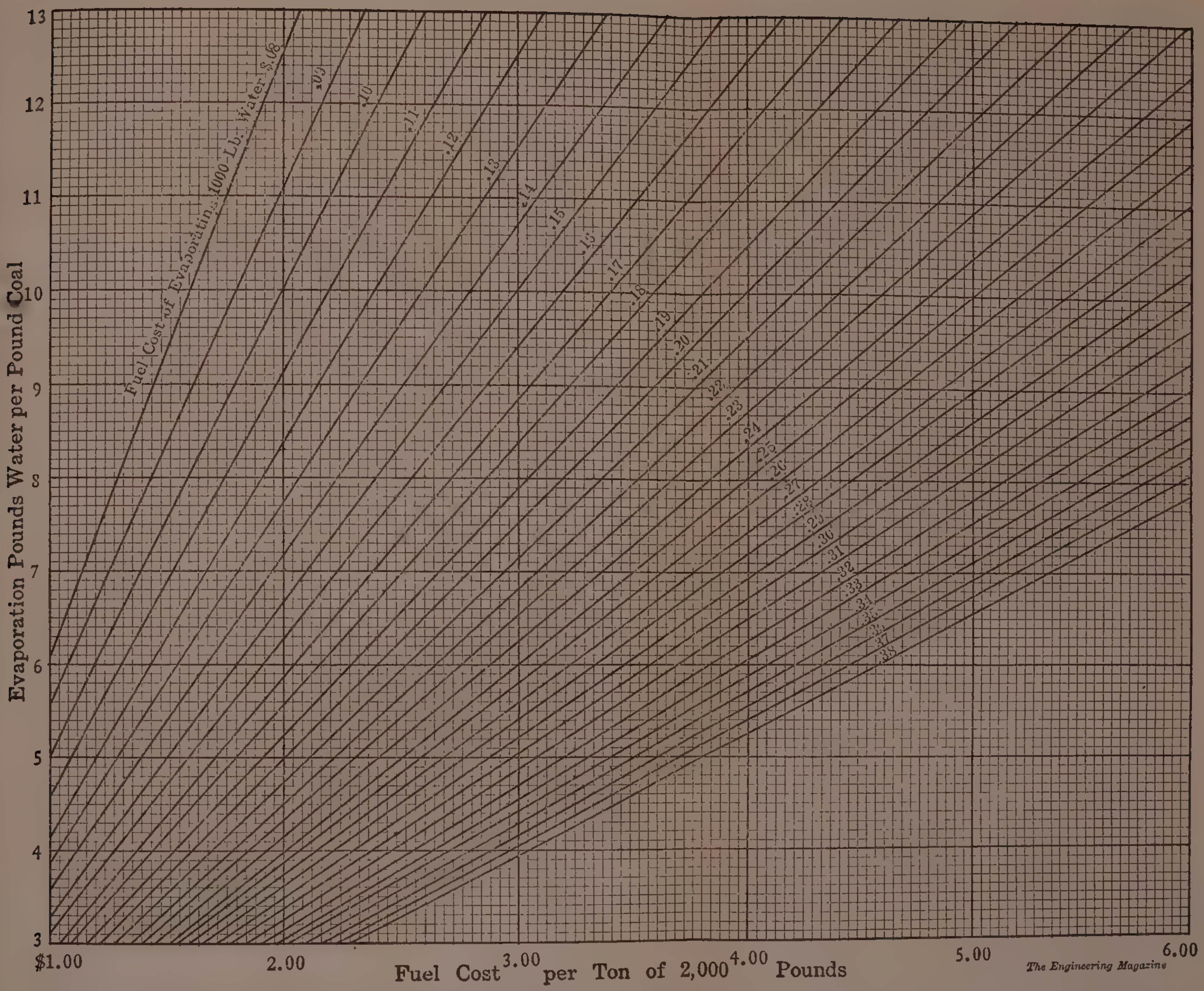


FIG. 18. DIAGRAM SHOWING COST OF EVAPORATING 1,000 LB. OF WATER AT DIFFERENT RATES OF EVAPORATION, AND PRICES OF FUEL

It is necessary to know only the cost of the coal per 2,000 pounds and the evaporation per pound in order to use the curve, which directly indicates the fuel cost of evaporating 1,000 pounds of water.

If the evaporation is taken as "actual" the result will be actual, or the equivalent evaporation may be used (equivalent evaporation = actual evaporation × factor of evaporation), in which case the curve will show the fuel cost of evaporating 1,000 pounds of water into steam from and at 212 degrees F.

It may be of interest to note that the efficiency system described on pages 239 to 241 inclusive is in successful operation. The set standard of efficiency is being maintained and the firemen are receiving their bonuses. Needless to say the results are equally gratifying to both the firm and firemen.

Chapter XI

BOILER TESTS

ALL the following tests were made in factory boiler plants, and the results and conditions are typical of the various kinds of boiler and furnace equipment indicated.

Tests on Bituminous Coal in Internally Fired Manning Type Upright Boilers

The following are records of three tests made on boilers in the same plant. The wide difference in the efficiencies of tests Nos. 1 and 3 is chargeable to operation alone. The term "operation" includes the cleanliness of the boiler heating-surfaces inside and out, the rate of driving, the skill of firing, all of which are under human control. These tests show the absolute necessity of keeping the factory boiler plant under a careful efficiency system so that it will at once become known when the performance falls off. Test No. 3 was made about three years after tests Nos. 1 and 2,

and it was not known how long the heavy loss shown by test No. 3 had been going on.

It is also to be noted that when this type of boiler is forced there is a decided tendency to increase the percentage of CO in the flue gases, and there is an equally strong tendency to increase the escape of unburned volatile hydrocarbons, especially immediately after firing.

The reason is that the gases pass very quickly from the furnace into the small fire-tubes of the boiler, with such speed that there is not sufficient time for their complete diffusion with the oxygen. Furthermore, the water-legged firebox reduces the temperature of the combustion space, which factor further retards complete combustion. When the fuel gases have once entered the tubes their temperature is so quickly reduced that further combustion is practically impossible.

A similar chilling effect takes place also on the standard type of *water-tube boiler* (unless a special furnace is provided) which makes this kind of boiler more of a smoke producer than one of horizontal tubular design. The water-tube boiler has the advantage over the vertical upright fire-tube boiler in the possession of a fire-brick furnace which helps to elevate and maintain the temperature of combustion.

EFFICIENCY TESTS ON UPRIGHT INTERNALLY FIRED BOILERS OF THE MANNING TYPE

	Test No. 1	*Test No. 2*	*Test No. 3*
Kind of fuel	Bituminous—Run-of-Mine		
Kind of furnace	Internally fired, shaking grates		
Kind of boiler	Manning Type—Upright		
Number of boilers tested	1	1	3
Method of starting and stopping test	alternate	alternate	alternate
Date of trial	4/20/09	4/15/09	3/21/12
Duration of trial	6 hrs., 2 min.	7.033 hrs.	8 hrs.
Dimensions and Proportions			
Grate surface—sq. ft.	38.5	38.5	115.5
Height of furnace	....	....	60.75 in.
Water-heating surface	1570 sq. ft.	1570 sq. ft.	5050 sq. ft.
Superheating surface	477 sq. ft.	477 sq. ft.	1431 sq. ft.
Ratio of water-heating surface to grate surface	41.7 to 1	41.7 to 1	43.7 to 1
Steam pressure by gauge	86.5 lb.	85 lb.	109 lb.

Force of draft between damper and boiler	0.34 in. w.g.	0.415 in. w.g.	0.535 in. w.g.
Average Temperature			
Of external air	....	....	26 degrees
Of fireroom	....	....	61.5 "
Of feed water entering boiler	*162 degrees*	*170 degrees*	*171 "*
Of escaping gases from boiler	*416 "*	*563 "*	*549 "*
Fuel			
Weight of coal as fired	2,774 lb.	6,771 lb.	26,698 lb.
Percentage of moisture in coal	3.24 per cent	5.06 per cent	3.88 per cent
Total weight of dry coal consumed	2,684 lb.	6,428 lb.	25,662 lb.
Percentage of ash and refuse in dry coal	12.3 per cent	12.8 per cent	....
Proximate Analysis of Coal			
Fixed carbon	....	....	61.02 per cent
Volatile matter	....	....	30.25 per cent
Moisture	....	....	3.88 per cent
Ash	....	....	8.73 per cent
Sulphur, separately determined	....	....	2.21 per cent

EFFICIENCY TESTS ON UPRIGHT INTERNALLY FIRED BOILERS OF THE MANNING TYPE
(Continued)

	Test No. 1	*Test No. 2*	*Test No. 3*
Fuel per Hour			
Dry coal consumed per hour	447 lb.	914 lb.	3,208 lb.
Combustible consumed per hour	....	....	2,927 lb.
Dry coal per sq. ft. of grate surface per hour	11.6 lb.	23.7 lb.	27.8 lb.
Combustible per sq. ft. of water-heating surface per hour	....	....	25.3 lb.
Calorific Value of Fuel			
Calorific value by oxygen calorimeter per lb. of dry coal	14,011 B.t.u.	13,034 B.t.u.	13,992 B.t.u.
Water			
Total weight of water fed to boiler	25,378 lb.	54,574 lb.	203,835 lb.
Factor of evaporation	1.0890	1.0802	1.0835

Equivalent water evaporated into steam from and at 212 degrees	27,637 lb.	58,951 lb.	220,855 lb.
Water per Hour			
Equivalent evaporation per hour from and at 212 degrees	4,583 lb.	8,385 lb.	27,607 lb.
Equivalent evaporation per hour from and at 212 degrees per sq. ft. of water-heating surface	2.92 lb.	5.34 lb.	5.46 lb.
Horse Power			
Horse power developed (34½ lb. of water evaporated per hour into dry steam from and at 212 degrees, equals 1 horse power)	*132.8*	*243*	*800*
Builders' rated horse power at 12 sq. ft. per horse power	130.7	130.7	420.4
Percentage of builders' rated horse power developed	*101.5 per cent*	*186 per cent*	*190.4 per cent*
Economic Results			
Water apparently evaporated under actual conditions per lb. of coal as fired	9.15 lb.	8.06 lb.	7.635 lb.

EFFICIENCY TESTS ON UPRIGHT INTERNALLY FIRED BOILERS OF THE MANNING TYPE
(*Continued*)

	Test No. 1	*Test No. 2*	*Test No. 3*
Economic Results—Continued			
Equivalent evaporation from and at 212 degrees per lb. of coal as fired	9.963 lb.	8.706 lb.	8.27 lb.
Equivalent evaporation from and at 212 degrees per lb. of dry coal	*10.3 lb.*	*9.17 lb.*	*8.606 lb.*
Equivalent evaporation from and at 212 degrees per lb. of combustible	....	....	9.43 lb.
Efficiency			
Efficiency of boiler, including the grate; heat absorbed by the boiler, per lb. of dry coal, divided by the heat value of 1 lb. of dry coal	*71.4 per cent*	*68.2 per cent*	*59.7 per cent*
Efficiency of boiler including grate, based on available heat in coal after deducting loss due to moisture in coal	71.5 per cent	69.6 per cent	59.8 per cent

Cost of Evaporation			
Cost of coal per ton of 2,240 lb. delivered in boiler room	$4.00	$4.00	$4.00
Cost of fuel for evaporating 1,000 lb. of water under actual observed conditions	*$0.195*	*$0.221*	*$0.234*
Cost of fuel used for evaporating 1,000 lb. of water from and at 212 degrees based on dry coal	*$0.173*	*$0.195*	*$0.208*
Methods of Firing			
Kind of firing (spreading, alternate, or coking)			spreading
Average intervals between firings for each furnace	11.25 min.	6.5 min.	8.8 min.
Analyses of Dry Gases			
Carbon dioxide	8.1	12.8	8.05
Oxygen (O)	11.9	4.7	10.08
Carbon monoxide (CO)	0.27	2.72	0.43
Hydrogen and hydrocarbons, nitrogen (by diff.) (N)			81.44

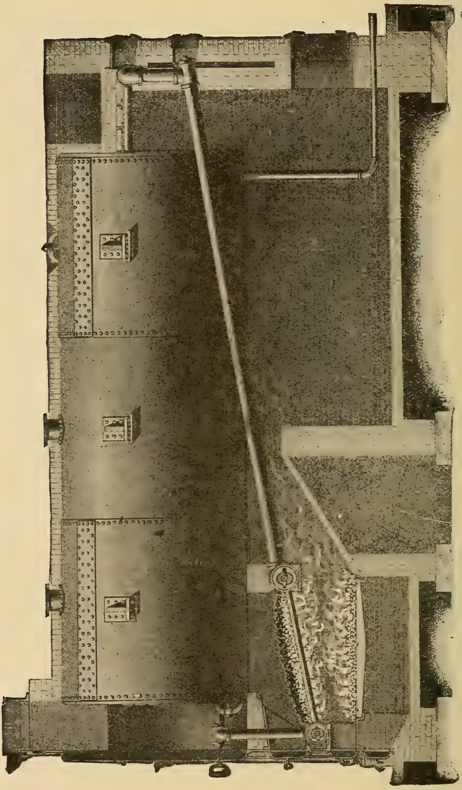


FIG. 19. THE HAWLEY DOWN DRAFT FURNACE

Courtesy of Hawley Down Draft Furnace Co.

The vertical upright has the counter advantage of lower radiation losses, and the best way (for a simple rule) to fire the Manning type is to produce a flame no longer than the height of the firebox, so that combustion is completed before the gases enter the tubes. This is possible by regulation of the air supply and by guarding against undue forcing. The flame length is approximately directly proportional to the degree of forcing (or rate of combustion) and inversely proportional to the air supplied per pound of fuel.

Forcing a fire makes more gas, which increases its velocity, thus drawing out the flame. Reducing the air supply makes a longer flame because the reduced supply of oxygen must be given a longer time to "find" and burn the combustible gases. Thus by the time the oxygen molecules have found their mates in the combustible gases, they have traveled a greater distance in the combustion chamber and the flame is longer. Where the flame stops, combustion stops also, whether it be complete or incomplete.

Tests on Bituminous Coal with Down-draft Grates and Horizontal Tubular Boilers

The following tests were made under regular daily operating conditions in a New

England factory, and the high efficiencies obtained were checked up over a long period of time by means of daily coal and water-weighing records which proved the performances herewith quoted to be average and not special test results. Figs. 19 and 20 show the type of furnace equipment and a detail of the water grate.

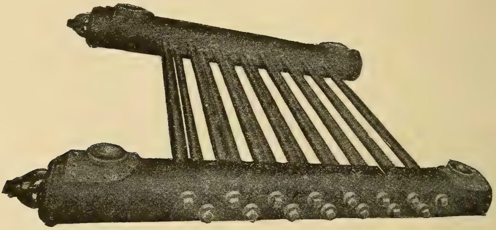


FIG. 20. HAWLEY WATER-TUBE GRATE

Showing plugs opposite tubes for cleansing purposes when necessary

The operation of the system consists in firing the green coal upon the upper grate, from which the combustion of the greatest part occurs. The burning takes place at the under surface of this bed of fuel which reaches a temperature of incandescence while the upper surface remains comparatively cool. The result is that the volatile portion of the fresh coal must pass downward (by the action of the draft) through the incandescent layer

which by intimate contact raises these fuel gases and air to a high combustion temperature. This process prevents the production of smoke and practically insures complete combustion. The CO gas from the top grate must also become heated in a similar manner.

As the coal becomes coked and crumbles, the smaller particles of coke fall through the wide spacing of the water grate upon the secondary grate with fine air spacing at the bottom, which has its own draft door underneath. This fire may therefore be separately regulated to make possible a correct adjustment of the air supply. The two fires mingle between the upper and lower grates, the water grate receiving its air supply through the firing door over the top of the green coal. Therefore since this door is open all the time, the great inrush of cold air which occurs when an ordinary furnace is fired is entirely eliminated.

This furnace of course adds a certain complication to the boiler itself due to the water grate and the tubes connecting it to the boiler, and the equipment must be scrupulously guarded to prevent the possibility of burning or sagging of the grates and tubes. These tests are of particular interest as an indication of the economy of combustion obtainable with this type of furnace.

TESTS ON BITUMINOUS COAL WITH HAWLEY TYPE OF DOWN-DRAFT GRATES AND HORIZONTAL TUBULAR BOILERS

	Test No. 1	*Test No. 2*
Kind of fuel	Bituminous, Run-of-Mine	
Kind of furnace	Hawley Down-draft	
Kind of boiler	Horizontal Tubular	
Method of starting and stopping test	Alternate	
Date of trial	6/3/09	6/4/09
Duration of trial	10 hrs. 4 min.	10 hrs. 1 min.
Dimensions and Proportions		
Grate surface—sq. ft	48	57.5
Grate surface of water grate—sq. ft	24	30
Grate surface of lower grate—sq. ft	24	27.5
Water-heating surface including heating surface of Hawley System	1,692 sq. ft.	1,821 sq. ft.
Superheating surface	0	0
Ratio of water-heating surface to total grate surface	35.3 to 1	31.7 to 1

Ratio of water-heating surface to grate surface of water grate	70.6 to 1	60.7 to 1
Width of draft openings in grate (lower)	½ in.	½ in.
Per cent draft area in grate (upper) projected	31 per cent	31 per cent
Per cent draft area in grate (lower)	45 per cent	45 per cent
Average Temperatures		
Steam pressure by gauge	102.5 lb.	109 lb.
Force of draft between damper and boiler	0.35 in.	0.456 in.
Force of draft in combustion chamber (back end)	0.30 in.	0.337 in.
Of feed water entering boiler	184.7 degrees	179.4 degrees
Of escaping gases from boiler	492 degrees	507 degrees
Of air entering fire box	117 degrees	120 degrees
Of external air	82 degrees	79 degrees
Fuel		
Weight of coal as fired	4,087 lb.	5,383 lb.
Percentage of moisture in coal	1.81 per cent	2.27 per cent
Total weight of dry coal consumed	4,013 lb.	5,261 lb.
Total ash and refuse dropped through grate and cleaned from grate surface	377 lb.	622 lb.
Percentage of ash and refuse in dry coal	9.394 per cent	11.82 per cent

TESTS ON BITUMINOUS COAL WITH HAWLEY TYPE OF DOWN-DRAFT GRATES AND HORIZONTAL TUBULAR BOILERS—*Continued*

	Test No. 1	*Test No. 2*
Fuel per Hour		
Coal as fired consumed per hour	406 lb.	537 lb.
Dry coal consumed per hour	339 lb.	525 lb.
Dry coal per sq. ft. of grate surface per hour figuring water grate only	16.9 lb.	17.5 lb.
Calorific Value of Fuel		
Calorific value per lb. of dry coal	13,830 B.t.u.	13,917 B.t.u.
Moisture in coal as delivered to fireman	1.81 per cent	2.27 per cent
Available heat in one lb. of coal as fired after deducting loss due to moisture and heat to evaporate the moisture	13,558 B.t.u.	13,574 B.t.u.
Water		
Total weight of water fed to boiler	39,755 lb.	53,458 lb.
Water actually evaporated	39,755 lb.	53,458 lb.

Factor of evaporation	1.0680	1.0753
Equivalent water evaporated into steam from and at 212 degrees	42,459 lb.	57,483 lb.
Water per Hour		
Water evaporated per hour	3,948 lb.	5,333 lb.
Equivalent evaporation per hour from and at 212 degrees	4,216 lb.	5,739 lb.
Equivalent evaporation per hour per sq. ft. of heating surface	2.485 lb.	3.15 lb.
Horse Power		
Total horse power developed	122.2	166.3
Builders' rated horse power at 12 sq. ft. per horse power (including heating surface of Hawley System)	141.3	151.5
Percentage of rated horse power developed	86.4 per cent	110 per cent
Economic Results		
Water apparently evaporated under actual conditions per lb. of coal as fired	9.727 lb.	9.931 lb.
Equivalent evaporation from and at 212 degrees per lb. of coal as fired	10.389 lb.	10.678 lb.

TESTS ON BITUMINOUS COAL WITH HAWLEY TYPE OF DOWN-DRAFT GRATES AND HORIZONTAL TUBULAR BOILERS—*Continued*

	Test No. 1	*Test No. 2*
Economic Results—Continued		
Equivalent evaporation from and at 212 degrees per lb. of dry coal	10.58 lb.	10.93 lb.
Efficiency		
Efficiency of the boiler including grate based on the available heat in the coal as fired	74.2 per cent	75.96 per cent
Efficiency of boiler including grate based on heat in dry coal	73.9 per cent	75.8 per cent
Cost of Evaporation		
Cost of coal per ton of 2,240 lb. delivered in boiler room	$3.75	$3.75
Cost of coal evaporating 1,000 lb. of water from and at 212 degrees	$0.1610	$0.1566

FLUE-GAS ANALYSIS MADE DURING TEST ON BOILER NO. 1

Time Sample Taken	CO_2	O	CO
9:20 a.m.—Fire doors shut, draft 0.25, bottom grate door slightly open, ash doors shut......	6.3	13.7	—
10:06 a.m.—Damper open 2/3, fire doors wide open, bottom grate doors cracked.............	8.2	10.6	0.4
12:45 a.m.—Regular running order, fire doors wide open, lower grate doors cracked, ash door slightly open, top of fire half coked......	7.5	12.8	0.1
1:53 p.m.—Same conditions except top of fire well coked..............................	8.2	11.2	—
Green fire on top, grate doors as usual.	6.5	13.5	0
Lower fire thin in spots...............	5.9	14.3	0
3:25 p.m.—Well coked on upper grate..............	10.0	11.0	0
3:50 p.m.—Half coked on upper grate..............	7.3	12.5	0.2
4:22 p.m.—Upper grate not fully coked............	8.2	11.2	0
5:56 p.m.—Well coked on upper grate..............	8.0	—	—
Average..............................	7.6	12.31	0.10

These samples were taken in the uptake and include whatever leakage of air may have occurred through the boiler setting, and they should be used with this fact in mind.

Since the horizontal tubular boiler, hand-fired with soft coal without any special equipment, constitutes the most used type of steam-generating apparatus in factory practice, the following test on such an arrangement will be of interest.

This setting was of a type commonly found, the principal exception being a return of the gases over the top of the boiler, a feature which is going out of practice on account of the possibility of dangerous overheating of this portion of the shell which, having only steam gas in contact with its inside surface, does not readily transmit the heat it receives.

The efficiency of 66.6 per cent, which may be considered fairly good for this equipment with unskilled handling, was due largely to the frequency of the firing tending to maintain a maximum furnace temperature with a consequent lowering of the losses due to unburned volatile and CO gases. The spreading system of firing, the most generally used method, was employed, but the intervals between firings were unusually short, being less than six minutes.

Barring the cooling effect of air entering the fire doors, it is correct to state that the more often and more lightly a furnace is fired the higher will be its temperature and combustion efficiency.

TEST ON BITUMINOUS SLACK COAL WITH AN ORDINARY HAND-FIRED HORIZONTAL TUBULAR BOILER WITH STATIONARY GRATES, SHOWING "GOOD" RESULTS FOR THE EXISTING CONDITIONS

Kind of fuel	Youghiogheny slack
Kind of furnace	Stationary grate
Kind of boiler	Horizontal tubular
Method of starting and stopping test	Alternate
Date of trial	May 2, 1907
Duration of trial	9 hours
Dimensions and Proportions	
Draft area in grate	40 per cent
Width of air spaces	½ in.
Height of furnace	30 in.
Water-heating surface	1,543 sq. ft.
Average Temperatures, Pressures, etc.	
Of external air	53 degrees
Of fireroom	64 degrees
Of feed water entering boiler in test	117 degrees
Of escaping gases from boiler	567 degrees
Steam pressure by gauge—lbs. per sq. in.	112 lb.
Force of draft between damper and boiler	0.60 in. w. g.
Force of draft in furnace	0.54 in. w. g.
Fuel	
Size and condition	Slack coal
Weight of coal as fired	5,800 lb.

TEST ON BITUMINOUS SLACK COAL—*Continued*

Fuel—Continued	
Percentage of moisture in coal.	3.01
Total weight of dry coal consumed....................	5,625 lb.
Total ash and refuse removed from above and below grates	679 lb.
Fuel per Hour	
Dry coal consumed per hour..	625 lb.
Calorific Value of Fuel	
Calorific value by oxygen calorimeter, per lb. of dry coal..	13,142 B.t.u.
Available heat in a lb. of coal as fired after deducting loss due to moisture in coal.....	12,710 B.t.u.
Water	
Total weight of water fed to boilers....................	44,341 lb.
Factor of evaporation........	1.409
Equivalent water evaporated into steam from and at 212 degrees..................	50,889 lb.
Water per Hour	
Equivalent evaporation per hour from and at 212 degrees	5,656 lb.
Equivalent evaporation per hour from and at 212 degrees per sq. ft. of water-heating surface..................	3.68 lb.
Horse Power	
Boiler horse power developed.	163.9

TEST ON BITUMINOUS SLACK COAL—*Continued*

Horse Power—Continued	
Builders' rated horse power at 12 sq. ft. per horse power...	128.5
Percentage of rated horse power developed...........	128.5 per cent
Economic Results	
Water apparently evaporated under actual conditions per lb. of coal as fired.........	7.645 lb.
Equivalent evaporation from and at 212 degrees per lb. of coal as fired...............	8.774 lb.
Equivalent evaporation from and at 212 degrees per lb. of dry coal..................	9.046 lb.
Efficiency	
Efficiency of boiler, including the grates; heat absorbed by the boiler, per lb. of coal as fired, divided by the available heat of 1 lb. of coal as fired after deducting loss due to moisture................	66.6 per cent
Method of Firing	
Kind of fire (spreading, alternate, or coking)............	spreading
Average intervals between firings......................	5.57 minutes

Another test on a hand-fired horizontal tubular boiler equipped with shaking grates but having no return flue over the top of the shell is of value, in general confirmation of what may be regularly obtained without special equipment but with fairly good firing which in the following case was by the alternate method.

It is interesting to note in the flue-gas analysis made during this test the feast-and-famine tendency of the air supply which is characteristic of hand firing. Thus immediately after firing, the air in proportion to the coal decreases, as indicated by the rise of CO_2, and just before firing the CO_2 is low, showing a greater excess of air.

Various patented appliances have been devised for the purpose of equalizing the air supply to the requirements at all stages of the fire. Such inventions are generally speaking a failure. They tend to flood the furnace with air far in excess of the true requirements, and are likely to lower the temperature of the furnace during the volatilization of hydrocarbons immediately after firing, just when a maximum temperature is wanted to ignite these gases.

TEST ON HORIZONTAL TUBULAR BOILER WITH SHAKING GRATE AND HAND-FIRED

Kind of fuel	Bit.-run-of-mine
Kind of furnace	Shaking grate, hand-fired
Kind of boiler	Hor. tubular
Method of starting and stopping test	Alternate
Date of trial	Feb. 7, 1911
Duration of trial	8 hrs.
Dimensions and Proportions	
Grate surface (7 ft. × 7 ft.)	49 sq. ft.
Air space	5/16 in.
Percentage of draft area	35.7 per cent
Water-heating surface	2,526 sq. ft.
Superheating surface	0 sq. ft.
Ratio of water-heating surface to grate surface	51.6 to 1
Average Temperatures, Pressures, etc.	
Steam pressure by gauge	86.5 lb.
Temperature of feed water entering boiler	175 degrees
Temperature of gases leaving boiler	580 degrees
Temperature of air entering ash-pit	68.5 degrees
Temperature of external air	28.2 degrees
Force of draft between damper and boiler	0.47 in. w. g.
Force of draft over fire	0.27 in. w. g.

TEST ON HORIZONTAL TUBULAR BOILER—*Continued*

Average Temperatures, Pressures, etc.—Continued	
Force of draft at rear of combustion chamber when tested (4 readings)	0.30 in. w. g.
Weather	Snow—damp air
Fuel	
Weight of coal as fired	9,095 lb.
Percentage of moisture in coal	5.07 per cent
Total weight of dry coal consumed	8,634 lb.
Fuel per Hour	
Dry coal consumed per hour	1,079 lb.
Dry coal per sq. ft. of grate surface per hour	22 lb.
Calorific Value of Fuel	
Calorific value per lb. of dry coal.	12,900 B.t.u.
Moisture in coal as delivered to fireman	5.07 per cent
Water	
Total weight of water fed to boiler	72,080 lb.
Factor of evaporation	1.0754
Equivalent water evaporated into steam from and at 212 degrees	77,514 lb.
Water per Hour	
Water evaporated per hour from and at 212 degrees	9,689 lb.
Water evaporated per hour from and at 212 degrees per sq. ft. of water-heating surface	3.84 lb

TEST ON HORIZONTAL TUBULAR BOILER—*Continued*

Horse Power	
Total horse power developed....	281
Builders' rated horse power at 12 sq. ft. per horse power	210
Percentage of builders' rated horse power developed.......	133.7 per cent
Economic Results	
Water apparently evaporated under actual conditions per lb. of coal as fired.................	7.92 lb.
Equivalent evaporation from and and at 212 degrees per lb. of coal as fired................	8.52 lb.
Equivalent evaporation from and at 212 degrees per lb. of dry coal.......................	8.98 lb.
Efficiency	
Efficiency of boiler including grate based on dry coal............	67.25 per cent
Efficiency of boiler including grate, based on available heat subtracting moisture loss.........	67.5 per cent
Cost of Evaporation	
Cost of coal per ton of 2,000 lb. delivered in boiler room.......	$2.40
Cost of coal for evaporating 1,000 lb. of water into steam from and at 212 degrees............	$0.141

TEST ON HORIZONTAL TUBULAR BOILER—(*Continued*)

FLUE-GAS ANALYSIS

Time Sample Taken	CO_2	*O*	*CO*
10:38 a.m.—Just after firing	11.5	....	....
11:40 a.m.—Just before firing	7.4	11.0	0
12:14 m.—Just before firing	8.5	10.8	0
1:11 p.m.—Just before firing	9.4	9.8	0
1:55 p.m.—Just after firing	13.0	5.0	0
3:09 p.m.—Short time before firing	4.3	15.7	0.3
3:55 p.m.—5 minutes before firing	7.2	12.5	0.5
4:55 p.m.—Just before raking	5.7	....	....

The above tests show an irregular supply of air to the fire, but always an excess.

The tests show very slight loss from CO (partially burned carbon) but the loss from uncombined or excess air is at times large.

Too much grate surface with a consequently low rate of combustion and an excessive air supply represents a commonly found condition which results in a heavy loss of fuel.

Such a case is illustrated in the test following. The two horizontal tubular boilers were operated at about one-half of their normal capacity. This brought into service double the amount of grate surface required for a normal consumption of fuel per square foot. The draft intensity was normally strong and the supply of coal was abnormally low, so that the air supply far exceeded the requirements of the coal as fired. The result was an average CO_2 of only a little over 5 per cent, and the low efficiency of 52.8 per cent of boilers and furnaces combined.

A peculiar circumstance connected with this test is also significant. The engineer was unable to keep up the standard steam pressure even with twice the boiler heating and grate surface that was really necessary, and he thought the trouble was due to weak draft. The real causes of inefficiency were those already noted, combined with infrequent firing, and when they were eliminated by carrying all the load on a single boiler with reduced grate surface and teaching him how to fire, the efficiency of the boiler and

TESTS SHOWING POOR RESULTS ON TWO ORDINARILY SET HORIZONTAL TUBULAR FACTORY BOILERS

	No. 1 Boiler	*No. 2 Boiler*
Kind of furnace	Ordinary	Shaking Grate "Yingling"
Kind of boiler	Hor. tubular	Hor. tubular
Kind of fuel	Bituminous	Bituminous
Method of starting and stopping test	alternate	alternate
Grate surface—each furnace 6 ft. by 6 ft	36 sq. ft.	36 sq. ft.
Water-heating surface	1,240 sq. ft.	1,304 sq. ft.
Superheating surface	0	0
Ratio of water-heating surface to grate surface	34.5	36.2
Width of draft openings in grate	3/8 in.	....
Per cent draft area in grate	43 per cent	....
Total Quantities		
Date of trial	June 27, 1913	
Duration of trial	8 hours, 10 minutes	
Weight of coal as fired—soft	3,990 lb.	

Percentage of moisture in coal	1.6 per cent
Total weight of dry coal consumed	3,926 lb.
Percentage of ash in dry coal by analysis	13.24 per cent
Water actually evaporated	24,283 lb.
Factor of evaporation	1.18
Equivalent water evaporated from and at 212 degrees	28,654 lb.
Hourly Quantities	
Coal as fired consumed per hour	488 lb.
Dry coal consumed per hour	480 lb.
Dry coal per sq. ft. of grate surface per hour	6.7 lb.
Equivalent evaporation per hour from and at 212 degrees	3,509 lb.
Equivalent evaporation per hour per sq. ft. of heating surface	1.38 lb.
Average Pressures, Temperatures, etc.	
Steam pressure by gauge	58.4 lb.
Temperature of feed water entering boiler	67.5 degrees
Temperature escaping gases from boiler	471 degrees | 401 degrees
Temperature air entering ash-pit	81 degrees

TESTS SHOWING POOR RESULTS ON TWO ORDINARILY SET HORIZONTAL TUBULAR FACTORY BOILERS—*Continued*

	No. 1 Boiler	*No. 2 Boiler*
Average Pressures, Temperatures, etc.—Cont'd		
Temperature outside air	81 degrees	
Force of draft between damper and boiler, water gauge, inches	0.32	0.32
Horse Power		
Total horse power developed	101.7	
Builders' rated horse power at 12 sq. ft. per horse power	211.8 or	
	103.3	108.5
Percentage rated horse power developed	48.1 per cent	
Economic Results		
Water apparently evaporated under actual conditions per lb. of coal as fired	6.086	

Equivalent evaporation from and at 212 degrees per lb. of coal as fired	*7.181 lb.*
Equivalent evaporation per lb. of dry coal	*7.30 lb.*
Equivalent evaporation per lb. of combustible (by chemical analysis)	*8.42 lb.*
Efficiency	
Calorific value of the dry coal per lb	13,400 B.t.u.
Moisture in coal as delivered to firemen	1.6 per cent
Calorific value per lb. of combustible (based on chemical analysis)	15,445 B.t.u.
Available heat in one lb. of coal as fired after deducting loss due to moisture and heat to evaporate the moisture at temperature of steam in boiler	13,168 B.t.u.
Efficiency of boiler including grates based on the available heat in the coal as fired, D. M. Myers' revised standard	52.91 per cent
Efficiency of boilers and grates based on dry coal, A.S.M.E. Code	*52.8 per cent*

TESTS SHOWING POOR RESULTS ON TWO ORDINARILY SET HORIZONTAL TUBULAR FACTORY BOILERS—*Continued*

	No. 1 Boiler	*No. 2 Boiler*
Cost of Evaporation		
Cost of coal delivered and unloaded per 2,000 lb.	$3.50	
Cost of coal for evaporating 1,000 lb. of water under observed conditions	$0.287	
Cost of coal for evaporating 1,000 lb. of water from and at 212 degrees	$0.244	
Flue-Gas Analysis—Average of Samples Taken		
CO_2 (Samples from No. 1 Boiler taken through a.m. and No. 2 Boiler in p.m. Number of CO_2 samples—No. 1 Boiler = 14; No. 2 Boiler = 17. Number of complete analysis —No. 1 Boiler = 1; No. 2 Boiler = 1.)	7.06	5.32

See separate table of gas analysis on following page.

FLUE-GAS ANALYSIS

No. 1 Boiler Test

Sample Notes	CO_2	O	CO
9:10 a.m. 5 minutes after raking—Total = 19.4....	9.6	9.8	0
12:43 p.m.— 5 minutes after raking..................	12.2	....	....
12:54 p.m.— 2 minutes after covering...............	10.3	....	....
1:00 p.m.— 8 minutes after covering...............	8.0	....	....
1:05 p.m.—13 minutes after covering...............	6.0	....	....
1:09 p.m.—17 minutes after covering...............	5.6	....	....
1:15 p.m.—23 minutes after covering...............	4.8	....	....
(2 minutes before next covering)			
1:20 p.m.— 3 minutes after covering...............	6.8	....	....
1:24 p.m.— 7 minutes after covering...............	6.4	....	....
1:30 p.m.—13 minutes after covering...............	5.8	....	....
1:35 p.m.—18 minutes after covering...............	5.0	....	....
1:44 p.m.—27 minutes after covering...............	4.4	....	....
1:51 p.m.—34 minutes after covering...............	3.6	....	....
2:09 p.m.— 3 minutes after covering...............	10.4	....	....
Average CO_2 — 7.06			

furnace jumped up to about 70 per cent. The immediate result was a saving of over 24 per cent of fuel and no difficulty in maintaining the desired steam pressure. In fact, the dampers were now kept not quite half-open.

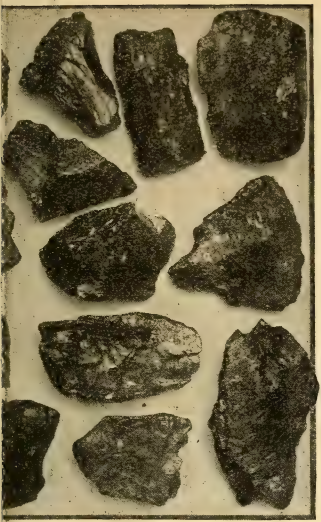
There was nothing unusual about the settings of these boilers, except that No. 2 had shaking grates.

Boiler Tests with Anthracite

In those districts where anthracite coal is available it is of essential value to know what performance may be expected with this fuel, especially in the smaller and cheaper sizes. For such information the following boiler tests made in factory plants are given. These will indicate the conditions surrounding both good and bad practice with Nos. 2 and 3 buckwheats together with a test on the higher priced No. 1 buckwheat made under very ordinary conditions.

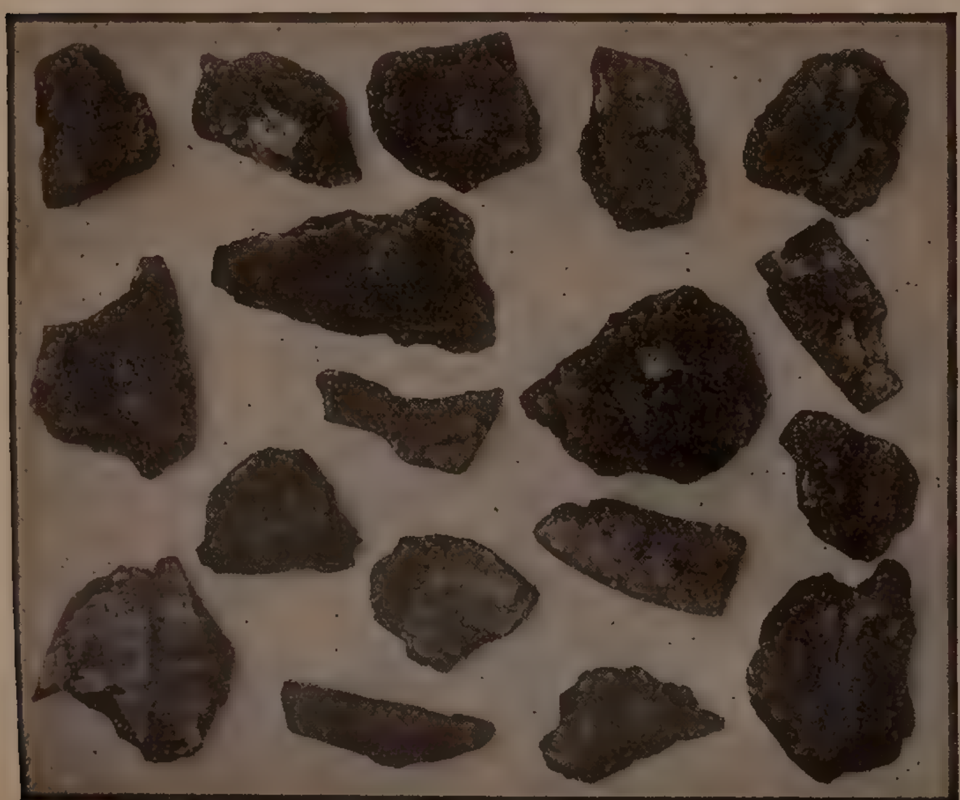
The standards governing the sizes of anthracite coals vary, but the schedule of sizes as recommended by the American Society of Mechanical Engineers, and as here illustrated in Fig. 21, may properly be accepted and used.

Two tests, given in detail on pages 282 to 290, were made in the same plant on No. 2 buckwheat mixed with a portion of soft coal.



PEA

Nº3 - BUCKWHEAT OR BARLEY



Nº1 - BUCKWHEAT

Nº4 - BUCKWHEAT OR CULM

Nº2 - BUCKWHEAT OR RICE

Standards for anthracite-coal sizes as recommended by the American Society of Mechanical Engineers and printed in the Society's *Journal*, November, 1912.

Name	Screen or Opening (Circular) Through or Over Which Coal Will Pass, Inches	
	Through	Over
Pea	$\frac{7}{8}$	$\frac{9}{16}$
No. 1 buckwheat	$\frac{9}{16}$	$\frac{5}{16}$
No. 2 buckwheat	$\frac{5}{16}$	$\frac{3}{16}$
No. 3 buckwheat	$\frac{3}{16}$	$\frac{3}{32}$
No. 4 buckwheat or culm	$\frac{3}{32}$	..

The sizes specified by the New York City Department of Water Supply, Gas and Electricity are practically the same as the above except for pea coal, which is specified as coal which will pass through $\frac{3}{4}$-inch and over $\frac{1}{2}$-inch openings.

Fig. 21. Actual Sizes of Anthracite Coal

Conditions of both tests, natural draft, common design of furnaces with B. & W. boilers. Both tests show poor results as to boiler capacity. Test A gave a good efficiency and test B a very low efficiency.

The conditions governing this wide variation in economic performance are readily obtainable from an analysis of the following records which are briefly discussed at their end.

The rating in test A was higher than in test B, which may be accounted for by the greater intensity of draft in the furnace, and by the shorter firing interval as recorded. Both ratings were low, as may be expected with a natural draft of normal intensity with this coal.

The great difference in efficiency (test A = 69.37 per cent and test B = 47.77 per cent) in favor of test A is principally due to the following conditions:

A firing interval of 16 minutes in test A (instead of 23½ minutes) operated to produce a more nearly uniform furnace temperature.

The draft area through the grate in test A was only 6.13 per cent, whereas the boilers of test B had grates with 40 per cent draft area, which largely accounts for the tremendous excess of air as denoted by the average of

Boiler Tests with Natural Draft, Fuel Principally Anthracite	*Test A* *No. 4 Boiler*	*Test B* *Nos. 1 and 2 Boilers*
Kind of fuel	No. 2 buckwheat and run-of-mine soft	
Kind of furnace	Regular B. and W.	
State of the weather	Clear and fine	Cloudy—good air
Date of trial	Oct. 3, 1911	Oct. 6, 1911
Duration of trial	9 hours	9 hours
Dimensions and Proportions		
Grate surface	112 sq. ft.	56 sq. ft. each
Approximate width of air spaces in grate	3/8 in. holes	1/4 in. slots
Proportion of air space to whole grate surface	6.13 per cent	40 per cent
Water-heating surface	5,061 sq. ft.	2,640 sq. ft. each
Ratio of water-heating surface to grate surface	45.2 to 1	47.2 to 1
Average Pressures		
Steam pressure by gauge	111 lb.	114 lb.
Force of draft between damper and boiler	0.50 in. w. g.	No. 1—0.35 No. 2—0.43

Force of draft in furnace	0.461 in. w. g.	No. 1—.33 in. No. 2—.36 in.
Average Tenperatures		
Of external air	60 degrees	66 degrees
Of air entering ash-pit	63 degrees	69 degrees
Of feed water entering heater	62 degrees	62 degrees
Of feed water entering boiler	185 degrees	174 degrees
Of escaping gases from boiler	431 degrees	No. 1—447 deg. No. 2—442 deg.
Fuel		
Size and condition	No. 2 buckwheat, hard and R.O.M. soft	
Weight of coal as fired—		
No. 2 buckwheat	82 per cent 9,672 lb.	80 per cent 9,600 lb.
Soft coal	18 per cent 2,128 lb.	20 per cent 2,400 lb.
Total weight of coal as fired	11,800 lb.	12,000 lb.
Percentage of moisture in coal—		
No. 2 buckwheat	7.38 per cent	8.23 per cent
Soft coal	4.18 per cent	3.34 per cent

Boiler Tests with Natural Draft, Fuel Principally Anthracite	*Test A* *No. 4 Boiler*	*Test B* *Nos. 1 and 2 Boilers*
Fuel—Continued		
Dry coal consumed—		
No. 2 buckwheat	8,958 lb.	8,810 lb.
Soft coal	2,039 lb.	2,320 lb.
Total weight of dry coal	10,997 lb.	11,130 lb.
Percentage of ash and refuse from grate at cleaning fire	9.78 per cent	5.43 per cent
Fuel per Hour		
Dry coal consumed per hour	1,222 lb.	1,236.5 lb.
Dry coal per sq. ft. of grate surface per hour	10.9 lb.	10.95 lb.
Calorific Value of Fuel		
Calorific value by oxygen calorimeter, per lb. of dry coal—		
No. 2 buckwheat	12,300 B.t.u.	12,438 B.t.u.
Soft coal	14,083 B.t.u.	14,055 B.t.u.
Calorific value of 1 lb. of dry mixture	12,632 B.t.u.	12,775 B.t.u.

Water		
Total weight of water evaporated	92,881 lb.	64,745 lb.
Factor of evaporation	1.0691	1.0812
Equivalent water evaporated into steam from and at 212 degrees	99,299 lb.	70,002 lb.
Water per Hour		
Equivalent evaporation per hour from and at 212 degrees	11,033 lb.	7,778 lb.
Equivalent evaporation per hour from and at 212 degrees per sq. ft. of water-heating surface	2.18 lb.	1.47 lb.
Horse Power		
Horse power developed (34½ lb. of water evaporated per hour into steam from and at 212 degrees, equals one horse power)	319.8	225.4
Builders' rated horse power (at 10 sq. ft. per horse power)	506.1	Both boilers 528
Percentage of builders' rated horse power developed	63.2 per cent	42.7 per cent

BOILER TESTS WITH NATURAL DRAFT, FUEL PRINCIPALLY ANTHRACITE	*Test A* *No. 4 Boiler*	*Test B* *Nos. 1 and 2 Boilers*
Economic Results		
Water apparently evaporated under actual conditions per lb. of coal as fired.	7.871 lb.	5,3954 lb.
Equivalent evaporation from and at 212 degrees per lb. of coal as fired	*8.415 lb.*	*5,8335 lb.*
Equivalent evaporation from and at 212 degrees per lb. of dry coal	*9.03 lb.*	*6.289 lb.*
Efficiency		
Efficiency of boiler, including the grate; heat absorbed by the boiler, per lb. of dry coal, divided by the *heat value of one lb. of dry coal*.	*69.37 per cent*	*47.77 per cent*
Cost of Evaporation		
Cost of coal per ton of 2,240 lb. delivered in boiler room.	No. 2 buckwheat $3.15 R.O.M. Soft $3.85	
Cost of mixture fired in test.	$3.28	$3.29

Cost of fuel for evaporating 1,000 lb. of water under observed conditions	$0.1856	$0.272
Cost of fuel used for evaporating 1,000 lb. of water from and at 212 degrees	*$0.1738*	*$0.252*
Methods of Firing		
Kind of firing (spreading, alternate, or coking)	spreading	spreading
Average thickness of fire	6 in. to 7 in.	6 in.
Average intervals between firings for each furnace	16 minutes	23½ minutes
Average interval between times of leveling or breaking up	Raked between firings	
Analyses of the Dry Gases		
Carbon dioxide (CO_2)	10.4 per cent	7.5 per cent
Oxygen (O) (by diff.)	10.1 per cent	13.0 per cent
Carbon monoxide (60)	none	none
Hydrogen and hydrocarbons and nitrogen (by diff.)	79.5 per cent	79.5 per cent

The flue-gas analysis accompanying this test is given on pages 288, 289, 290.

FLUE-GAS ANALYSIS

Taken During Test on Boiler No. 4

Boiler Tests with Natural Draft, Fuel Principally Anthracite	Per cent		
Sample Notes	CO_2	*O*	*CO*
1. Fired 8:30 a.m.—ash doors shut, boiler almost popping off, sample 8:40 a.m., 10 minutes after firing	11.8	8.2	0
2. Fired 9:10 a.m.—ash doors ½ open, sample 9:16 a.m., 6 minutes after firing	12.8	6.4	0
3. Fired 10:00 a.m.—ash doors ¼ open, sample 10:09 a.m., 9 minutes after firing	9.6	....	....
4. Fired 10:39 a.m.—sample 10:45 a.m., 6 minutes after firing	8.6	....	....
5. Sample 11 a.m., 5 minutes before starting to clean fire	6.0	....	....
6. Finished cleaning at 11:40 a.m., taking 35 minutes to clean	....	....	....
7. Sample 12:32 p.m., ash doors shut	10.3	....	....

8. Sample 1:23 p.m., 2 minutes before firing	10.0	....	....
9. Sample 1:44 p.m., 4 minutes before firing, ash doors ¼ open	11.0	....	....
10. Fired 2:06 p.m.—sample 2:08 p.m., 2 minutes after firing	10.8	....	....
11. Sample 2:40 p.m., ash doors shut	9.2	....	....
12. Sample 3:26 p.m., ash doors ½ open	12.0	....	....
13. Fired 4:05 p.m.—sample 4:07 p.m., 2 minutes after firing, ash doors ½ open	12.6	....	....
Average	10.4	7.3	0

NOTE.—In making boiler tests for commercial purposes it is desirable that only such tests be made as will have a likelihood of discovering sufficient preventable waste to pay large dividends on the expense involved. In many cases therefore it is advisable to concentrate the work of flue-gas analysis upon the CO_2 values, but this is done only after tests of both O and CO are made in conjunction with several of the highest percentages of CO_2 discovered. For instance, if with 11 per cent to 13 per cent CO_2 no CO is found, it is safe to assume for practical purposes that no CO will be found when the CO_2 runs considerably under these highest figures. This method economizes time and permits the record of a great many more CO_2 tests than would otherwise be possible, and yet as far as any work of improving the efficiency of the boilers is concerned all essential data are obtained.

FLUE-GAS ANALYSIS TAKEN DURING TEST ON BOILERS NOS. 1 AND 2

	Sample Notes	CO_2 *per cent*
1.	Sample 8:56 a.m.—Boiler No. 1	6.6
2.	Sample 9:20 a.m.—Boiler No. 2, ash doors all open	6.6
3.	Sample 9:53 a.m.—Boiler No. 1, 12 minutes before cleaning	8.6
4.	Sample 10:42 a.m.—Boiler No. 2, 12 minutes after firing	4.0
5.	Sample 11:13 a.m.—Boiler No. 2, ash doors ½ open	6.0
6.	Fired 11:50 a.m.—Boiler No. 1, sample 3 minutes after firing	11.0
7.	Sample 12:43 a.m.—Boiler No. 1, ashpit closed tight	5.0
8.	Fired 1:22 p.m.—Boiler No. 2, sample 1:25 p.m., 3 minutes after firing	7.0
9.	Fired 1:45 p.m.—Boiler No. 2, sample 3 minutes after firing	9.5
10.	Fired 2:00 p.m.—Boiler No. 1, ash doors ½ open, 7 minutes after firing	7.2
11.	Fired 2:25 p.m.—Boiler No. 1, 5 minutes after firing	7.8
12.	Fired 2:48 p.m.—Boiler No. 2, ash doors ½ open, 17 minutes after firing	8.1
13.	Sample 3:29 p.m.—Boiler No. 2, ash doors ½ open, just before raking	5.8
14.	Fired 3:44 p.m.—Boiler No. 1, sample 3:52 p.m., ash doors ½ open	7.0
15.	Raked 4:15 p.m.—Boiler No. 1, sample 6 minutes after raking	9.0
16.	Sample 4:49 p.m.—Boiler No. 2, 7 minutes after firing	7.0
	Average per cent CO_2	7.5

7.5 per cent CO_2 as compared to 10.4 CO_2 for test A.

The buckwheat coals *may* be burned with good results under natural draft as indicated by test A, but the horse-power output of the boiler will be below normal rating with chimneys of average height. When the chimney is of a height to give an intensity of draught sufficient for high ratings, the usual result is a very large excess of air in the furnace with a consequently low efficiency.

For burning the buckwheats the volume of air required is less per pound than for soft coals, but the intensity or pressure of draft must be greater in order to penetrate the more resistive bed of fuel. When a high stack is used to obtain the penetration, the stack damper must be opened wide to secure the maximum intensity at the furnace. This also increases the *volume,* which is not desired. Of course this ill effect can be counteracted by regulating the volume of air by means of the ashpit doors and leaving the stack and uptake dampers wide open to secure the desired intensity. But this requires very skillful operation as well as a high chimney, and furthermore the draft conditions change with the weather.

Consequently the most surely satisfactory method of burning the fine sizes of anthracite

is by the employment of mechanical or steam draft arranged to produce a pressure of air under the grate. A chimney of moderate height is needed to carry off the gases, but need be of no such height as would be required with natural draft alone. With this combination it is readily practicable to obtain separate regulation of the *intensity* of draft (for penetration) and the *volume* of draft for correct air supply. Changing the speed of the fan varies the intensity to any desired pressure, and the damper in the uptake is independently throttled down to reduce the volume of air to a minimum for good combustion.

It is of interest to note at this point that in New York city, where the small sizes of anthracite constitute the most used fuel, very great losses exist owing to surplus air in the fires, while losses due to incomplete combustion of the coal are utterly insignificant in comparison. The cure is less air, and the surest way to supply the less amount of air is to install forced draft. This sounds like a paradoxical statement, but in this case forced draft means the ability readily to control both the volume of air and the intensity of draft and to control them separately and independently. The higher the chimney which furnishes the draft for burning buckwheat

coals, the larger will be the probable saving in fuel that can be made by installing forced (and controlled) draft. I use the term "forced draft" as applying to any means of creating a pressure of air under the grate of the furnace, and in contra-distinction to "induced draft", in which case a fan is employed to exhaust the air from the boiler uptake. Induced draft is open to the same objections as natural chimney draft for burning the buckwheats since it operates and is controlled in the same manner.

Forced draft may be produced either by means of fans of various types, or by the use of steam jets which induce a current of air through a tube communicating inwardly to the ashpit of the furnace. This latter constitutes the simplest form of forced draft. In the past its principal drawback was its very large consumption of steam as compared to the amount required to operate a fan. But in recent years scientific design of steam jets has largely overcome this objection with the best types of steam blowers, and in small installations they will, with their added advantage of simplicity, often compete with fan-draft designs. In such cases the greatest drawback to the steam-jet system is the noise, or the roar of the escaping steam which may be disconcerting to the firemen

and hard on one's nerves. This may be partly mitigated by methods of muffling.

The following acceptance test was made on a steam-jet system burning straight No. 3 buckwheat and a very thorough test was made to determine the steam used by the blowers. This was found to be only 2.35 per cent of the boiler output when operating at 111.5 per cent of rated capacity (boiler rated at 12 sq. ft. per horse power). When purchasing a system of this kind specifications should be drawn to cover *net* efficiency based on the useful steam generated after deduction of the steam consumed by the blower.

For all *usual* installations of buckwheat-burning systems a possible under-grate pressure of 1½ inches of water should be provided, though for special cases where fires cannot be cleaned for long periods the resistance to air passage builds up as the fire bed increases in thickness and a much greater pressure is required.

In ordinary factory installations about ½-inch water pressure is sufficient with a clean fire, and this is gradually increased as the ash accumulates until just before cleaning it is usual to require between 1-inch and 1½-inch water pressure in the ashpit.

Except in extraordinary cases the fan (or steam jets) overcomes only that part of the

total draft resistance which is caused by the grate and the fire bed, so that atmospheric pressure or, more usually, a slight vacuum exists over the burning fuel. As a fireman would express it, "the fan shoves the draft through the grate and the fuel. Then the chimney picks it up and does the rest of the work. There is a push under the fire and a pull over it."

There should be no pressure in the furnace or the flames will blow out of the fire doors when opened. Also there should not be a strong vacuum, or cold air will rush in to cool the furnace while firing.

TEST ON NO. 3 BUCKWHEAT COAL WITH STEAM-INDUCED FORCED DRAFT, ON A HORIZONTAL TUBULAR BOILER. COMBUSTION ARCH BACK OF THE BRIDGE WALL

This test is typical of good results on No. 3 buckwheat coal in a well-designed furnace. Steam used for operation of draft—2.35 per cent of boiler output.

Kind of furnace, boiler set 4 feet above grate, combustion arch in front of blow-off	XY System
Kind of boiler, 6 ft. x 18 ft.-124 3 -in. tubes	Horizontal tubular
Kind of fuel	No. 3 buckwheat anthracite
Method of starting and stopping test	Alternate

TEST ON NO. 3 BUCKWHEAT COAL—*Continued*

Grate surface, sq. ft	42 sq. ft. (6 ft. × 7 ft.)
Water-heating surface, sq. ft	1,877 sq. ft.
Ratio of water-heating surface to grate surface	44.7 to 1
Width of draft openings in grate	5/16-in. round holes
Per cent draft area in grate	8.5
Total Quantities	
Date of trial	Dec. 13, 1912
Duration of trial, hours	8
Weight of coal as fired, lbs	5,907
Percentage of moisture in coal	9.58
Total weight of dry coal consumed	5,341 lb.
Total ash and refuse cleaned from top of grate	726 lb.
Percentage of same compared to dry coal burned	13.6
Water actually evaporated	41,358 lb.
Water used by blowers	974 lb.
Useful net actual evaporation	40,384 lb.
Factor of evaporation	1.19
Equivalent useful water evaporated from and at 212 degrees	48,057 lb.
Hourly Quantities	
Coal as fired consumed per hour	738 lb.
Dry coal consumed per hour	668 lb.
Dry coal per sq. ft. of grate surface per hour	15.9 lb.
Total water actually evaporated per hour	5,170 lb.

TEST ON NO. 3 BUCKWHEAT COAL—*Continued*

Hourly Quantities—Continued	
Total equivalent evap. per hr. per sq. ft. of heating surface	3.26 lb.
Equivalent evap. per hr. used by the two No. 7E blowers..	145 lb.
Total equivalent evaporation per hr. from and at 212 degrees....................	6,152 lb.
Average Pressures, Temperatures, etc.	
Steam pressure by gauge.....	112 lb.
Temperature of feed water entering boiler...............	67 degrees
Temperature escaping gases from boiler................	493 degrees
Temperature air entering ashpit......................	40 degrees
Force of draft between damper and boiler, water gauge, inches...................	0.273 vacuum
Draft pressure under grate, inches water..............	0.415 pressure
Draft pressure over grate.....	0.15 vacuum
Horse Power	
Total horse power developed..	178.3
Horse power developed—corrected for steam used by the blowers......................	*174.1*
Builders' rated horse power at 12 sq. ft. per horse power	156.0
Percentage rated useful horse power developed, useful steam	*111.5 per cent*
Percentage of horse power used by the blowers..............	*2.35 per cent*

TEST ON NO. 3 BUCKWHEAT COAL—*Continued*

Horse Power—Continued	
Boiler horse power used by the blowers	*4.2*
Economic Results	
Water apparently evaporated under actual conditions per lb. of coal as fired	7.01 lb.
Water actually evaporated corrected for steam used by blowers, per lb. of coal as fired	6.84 lb.
Equivalent useful evaporation per lb. of coal as fired	8.13 lb.
Equivalent evaporation per lb. of dry coal corrected for steam consumption of blower	*9.00 lb.*
Efficiency	
Calorific value of the dry coal per lb.	12,670 B.t.u.
Moisture in coal as delivered to firemen	9.58 per cent
Net efficiency of boiler including grate based on dry coal	*69 per cent*
Cost of Evaporation	
Cost of coal delivered at plant	$1.80 for 2,240 lb.
Cost of coal for evaporating 1,000 lb. of water from and at 212 degrees	*$0.0988*
Cost of coal for evaporating 1,000 lb. of water from and at 212 degrees, adding 10 cents per ton to include unloading and handling to fireroom	$0.1043

NOTES ADDED TO BOILER TEST

Tests Made on Percentage of CO_2 (Carbon Dioxide)	*per cent*
Sample taken at 2 p.m., about 3 minutes after firing, blast on	15
Sample taken at 2:20, about 5 minutes after firing, blast on	14
Sample taken at 2:45, fire well burned, blast on	11.8
Sample taken at 3:02, about 2 minutes after firing, blast on	11
Sample taken at 3:20, just after leveling fire, blast on	10

The above analyses show a good grade of combustion. The CO_2 should approach 15 per cent as nearly as possible for the best factory work and should not go below 10 per cent with No. 3 buckwheat or other hard coals. When the CO_2 drops below 10 per cent too much air is being admitted to the fire, and the damper should be throttled down to prevent this excess and the holes in fire bed carefully covered, and also a little greater depth of fire will tend to work towards reducing this excess air.

NOTES ON FIRING

The fire was cleaned at 8 a. m. The cleaning was completed and fire freshly covered with coal by 8:30, when the test was started. The fire was cleaned at noon, the noon hour being thirty minutes at this plant. The boiler was allowed to blow off twenty to

twenty-five minutes during this noon cleaning operation.

The fire was finally cleaned at 4 p. m. and the cleaning was completed and fire freshly covered with coal by 4:30 p. m. when the test was stopped.

There was, therefore, the same amount of unburned coal on the grate at the end as at the beginning of the test so that the amount fired between 8:30 and 4:30 was the amount which was actually burned during the test and charged against the boiler and the steam produced.

FIRING

The spreading method of firing was used throughout the test, and there was usually only a small interval of time between the covering of one side of the furnace and the covering of the other side. From 8:30 to 12:50, or about four hours and a half, there were 19 coverings of coal including the cleaning. Between 12:50 and 4:30, or about three hours and a half, there were about the same number of coverings. The firing during the first period mentioned was done by the expert fireman of the XY Mfg. Co., and in the afternoon period the fire was handled by the regular factory fireman with the exception of the final cleaning which was done by the XY

man. It is my judgment that the regular fireman kept as good a fire as the XY man; and there is no reason why the efficiency shown in this test should not be kept up continuously.

The following test on No. 1 buckwheat was made in the same plant as the test on No. 3 buckwheat immediately preceding. The plant originally burned the more expensive coal with natural draft and was then "put over" on the No. 3 buckwheat with forced draft. A comparison of the costs of evaporation of the respective systems will indicate the percentage of saving effected by this change. The advisability of such a change depends upon efficiencies and coal prices, dependability of coal supply, and upon the human factor. This latter may never be disregarded with impunity in changing over from one kind of coal to another which involves new firing methods.

TEST ON NO. 1 BUCKWHEAT; ANTHRACITE COAL, HAND FIRED

Horizontal tubular boiler, natural draft, combustion baffle arch under boiler.

Kind of fuel	No. 1 Buckwheat Anthracite
Kind of furnace	Stationary grate, special combustion arch under boiler

TEST ON NO. 1 BUCKWHEAT—*Continued*

Kind of boiler	Horizontal tubular
State of the weather	Clear and cold
Method of starting and stopping test	Alternate
Date of trial	12/18/13
Duration of trial	6 hrs., 4 min.
Dimensions and Proportions	
Grate surface	6 ft.x 6 ft. =36 sq. ft.
Height of furnace	36 in.
Approximate width of air spaces in grate	1/4 in. to 5/16 in.
Proportion of air space to whole grate surface	About 33 per cent
Dimensions of boiler	6 ft.x17 ft. with 82 4 in. tubes
Water heating surface	1,598 sq. ft.
Superheating surface	....
Ration of water-heating surface to grate surface	44.4 to 1
Steam pressure by gauge, average	39 lb.
Force of draft between damper and boiler	0.42 in. water
Force of draft in furnace	0.32 in. water
Force of draft or blast in ashpit	Atmospheric
Average Temperatures	
Of external air	31 degrees
Of fireroom	68 degrees
Of feed water entering boiler	59 degrees
Of escaping gases from boiler	542 degrees
Fuel	
Size and condition	Clean but wet
Weight of coal as fired	3,905 lb.
Percentage of moisture in coal	6.98 per cent

TEST ON NO. 1 BUCKWHEAT—*Continued*

Fuel—Continued	
Total weight of dry coal consumed	3,632 lb.
Total ash and refuse	650 lb. (approx.)
Fuel per Hour	
Dry coal consumed per hour	599 lb.
Dry coal per sq. ft. of grate surface per hour	16.6 lb.
Calorific Value of Fuel	
Calorific value by oxygen calorimeter, per lb. of dry coal	12,285 B.t.u.
Water	
Total weight of water fed to boiler	23,920 lb.
Factor of evaporation	1.1833
Equivalent water evaporated into steam from and at 212 degrees	28,305 lb.
Water per Hour	
Actual water evaporated per hour	3,936 lb.
Equivalent evaporation per hour from and at 212 degrees	4,660 lb.
Equivalent evaporation per hour from and at 212 degrees per sq. ft. of water-heating surface	2.91 lb.
Horse Power	
Boiler horse power developed	*135*
Builders' rated horse power at 10 sq. ft. per horse power	160
Percentage of builders' rated horse power developed	*84.4 per cent*

TEST ON NO. 1 BUCKWHEAT—*Continued*

Economic Results	
Water apparently evaporated under actual conditions per lb. of coal as fired..........	6.12 lb.
Equivalent evaporation from and at 212 degrees per lb. of coal as fired...............	7.25 lb.
Equivalent evaporation from and at 212 degrees per lb. of dry coal	*7.793 lb.*
Efficiency	
Efficiency of boiler, including the grate; heat absorbed by the boiler per lb. of dry coal, divided by the heat value of 1 lb. of dry coal (A. S. M. E. Code)	*61.5 per cent*
Cost of Evaporation	
Cost of coal per ton of 2,240 lb. delivered in boiler room....	\$2.75
Cost of fuel for evaporating 1,000 lb. of water under observed conditions..........	\$0.207
Cost of fuel used for evaporating 1,000 lb. of water from and at 212 degrees................	*\$0.1693*
Method of Firing	
Kind of firing (spreading, alternate, or coking)............	Spreading
Average thickness of fire......	About 5 in.
Average intervals between firings for each furnace.......	16.6 min.
Average interval between times of leveling or breaking up...	2 hrs.
Analysis of the Dry Gases	
Carbon dioxide (CO_2)........	10.3 per cent (average)

Methods for Boiler Testing

The following description of the author's method of weighing the feed water in a boiler test is herewith presented in connection with Fig. 22. An extremely accurate water weigher is used which automatically records the actual weight of the water fed to the boilers. By actual test this weigher shows a maximum error of one-third of one per cent and is far superior to hand-operated water tanks for accurate boiler testing.

The water weigher as shown at the right in Fig. 22, on page 306, is set above the sump tank, which may take the form of a hogshead or any kind of a tank of suitable size. Either hot or cold water under pressure is piped to the water weigher, its admission being regulated by a hand valve operable from the floor.

A separate feed pump, as shown, draws the water from the sump tank and delivers it through a special connection to the boiler or boilers to be tested. The diagram shows this special connection in its proper location for testing all the boilers in the battery together. When it is desired to test a single boiler, the connection shown at *A* is inserted in the individual branch of the feed line to any boiler as designated at *A-1*.

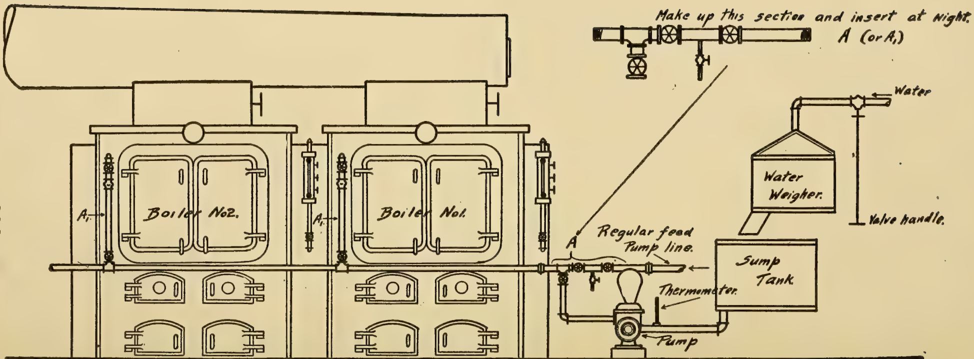


FIG. 22. WATER CONNECTIONS FOR BOILER TEST

In case of accident to test apparatus, regular feed pump can be used.

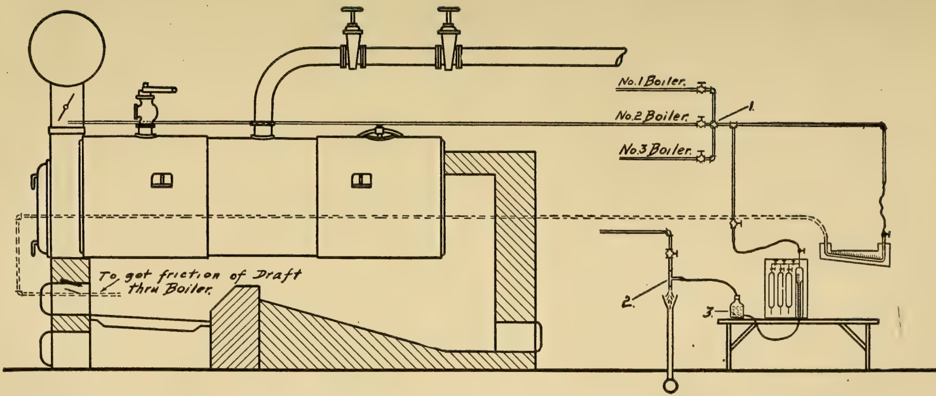


FIG. 23. FLUE-GAS AND DRAFT CONNECTIONS

For description see page 308

This special connection consists simply of a tee, three valves, and a drip connection with proper nipples as shown in the detail of Fig. 22.

One purpose of using this special piping arrangement is to allow quick starting and stopping of a boiler test. That is to say, by operating the valves of the special connection the boilers may be changed over instantly from the testing feed pump to the regular feed pump, and *vice versa.* This piping arrangement also shows up any leakage that might occur either from the boilers back through the regular feed pump while the test is on, or from the regular feed pump into the boilers during such period. The leak in either direction will be indicated by the drip connection as shown between the two valves on the horizontal run, the cock or valve being left open during the boiler test. One excellent feature of this standard connection is that it may be cut to accurate length and inserted in half an hour at noon-time or at night without disturbing the regular operation of the plant.

A convenient location for the feed-water thermometer is indicated on the suction of the test pump. This thermometer is of special construction and is very accurately calibrated.

Flue Gas and Draft Connections for Boiler Testing

This diagram (Fig. 23) shows the author's method for obtaining samples of flue gas for analysis from any boiler in the battery being tested.

This method makes double use of the sample piping from the uptake of each boiler. First this piping connects to the draft gage, for measuring the intensity of draft; and second, it furnishes a practically continuous sample of flue gas for analysis in the Orsat machine which stands on the table. A 1/8-inch standard gas pipe is run from the uptake of each boiler and connects to a cross or manifold at the point *1* close by the table. Thus the uptake of any boiler may be connected to either the draft gage or the flue-gas machine as desired.

The gas is drawn through the burette of the Orsat machine by means of a water operated ejector *2* which is connected to the top of the leveling bottle *3*. By breaking the connection between the water bottle and the ejector a sample of gas is entrapped and water-sealed in the burette of the Orsat machine. The dotted line shows a piping connection between the zero end of the Ellison draft gage and the furnace of the boiler. By its

use the frictional loss of draft between the furnace and the boiler uptake may be obtained on the draft gage in the form of a direct reading without calculation.

Chapter XII

COMBUSTION

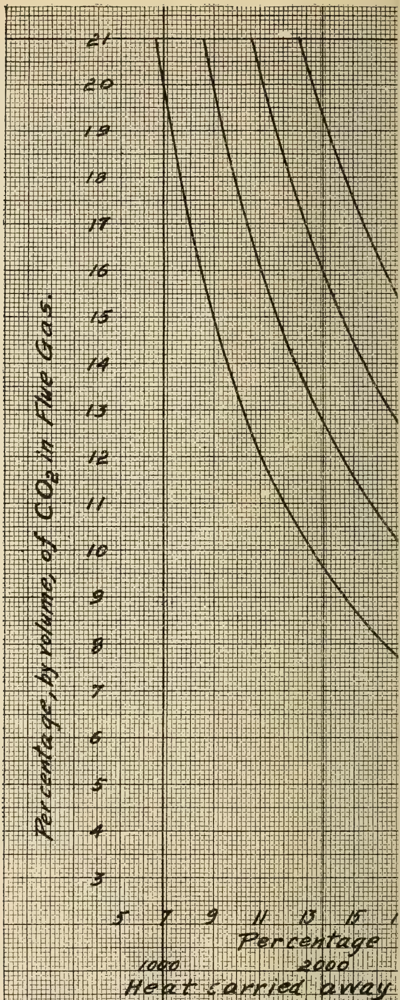
THE object of this chapter is to set forth in a convenient form the fundamental physical and chemical data relating to the combustion of fuel and to point out certain errors which are prevalent in connection with the use of combustion analysis for the determination of chimney losses. So much misunderstanding exists even among engineers in relation to the problems here involved that the thorough study given to this chapter is rendered well worth the effort it has required.

The one greatest loss in steam-boiler operation is that due to the heat carried away by the dry chimney gases. This loss (L) is equal to the weight of the dry products of combustion multiplied by their average specific heat and by the difference in temperature between the air in the fireroom and the escaping gases from the boiler.

Disregarding sulphur, carbon and hydrogen are the only heat-producing elements in a coal, and of these carbon alone produces dry gases upon combustion. The hydrogen burns to H_2O and together with the water contained in the coal produces the "wet" portion of the products of combustion, and the loss due to these constituents may best be calculated separately. Furthermore, the hydrogen constituent of a coal rarely exceeds 23 per cent of the heat value of the coal. We are therefore first concerned with the combustion of the carbon and its attendant chimney loss. As will be shown later in this chapter, a chimney-loss curve constructed on the basis of burning pure carbon and relating this loss to volumetric CO_2 as determined by an Orsat gas apparatus does not hold good for burning coal. This fact seems to be almost universally disregarded by the great majority of "combustion engineers."

Our first duty, however, is to construct such a pure-carbon curve as carefully as possible, and then to construct a curve for an actual coal in order in the course of the argument to develop quantitatively the differences that occur.

It is most common practice to find pure-carbon CO_2 curves applied for the determination of chimney losses. The results thus


Percentage, by volume, of CO2 in Flue Gas.
21
20
19
18
17
16
15
14
13
12
11
10
9
8
7
6
5
4
3
5 7 9 11 13 15
Percentage
1000 2000
Heat carried away
Based on Boiler-Room Tempc

obtained will be erroneous because we burn, not pure carbon, but coal which contains other constituents. In addition to this, even if we did have pure-carbon fuel many persons would obtain wrong deductions from the curve, for the reason that the losses of the curve are based not on the total carbon fed to the furnace but on that part only of the carbon which is burned and gasified. Furthermore, in developing the pure-carbon curve shown in Fig. 24, I have found serious errors in a set of curves which were looked upon as standard.

We will now proceed to consider the complete combustion[1] of a pound of carbon from a furnace grate. In the following discussion the nomenclature given in the explanatory table on pages 314 and 315 will be employed. These symbols for convenience have been made identical with those used by Edward A. Uehling in his valuable paper on "Combustion and Boiler Efficiency" presented before the American Society of Mechanical Engineers in 1910. Some of his method of reasoning has been employed in the present discussion.

---

[1] "Complete combustion" means entire oxidation of all the fuel constituents. Thus C must burn to CO_2 and H to H_2O. "Perfect combustion" should be defined as complete combustion with zero excess oxygen or air. The term "chemical combustion" may be used interchangeably with "perfect combustion".

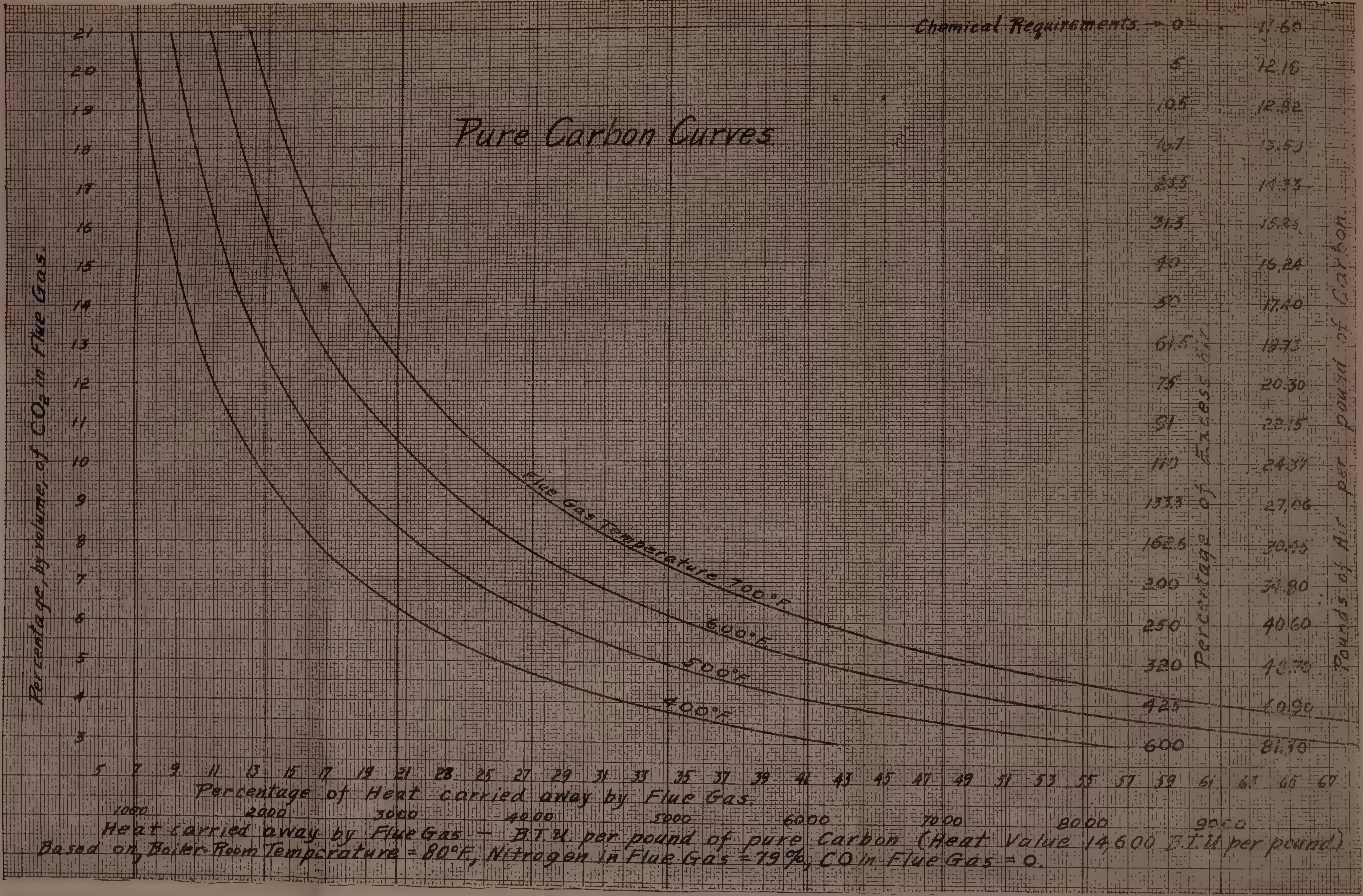


Fig. 24. Pure-Carbon Curve

L_d = Heat in B.t.u. carried away by dry chimney gases per weight of fuel containing one pound of carbon burned.

W_d = Pounds of dry chimney gases per weight of fuel containing one pound of carbon.

L = Pounds of dry chimney gases per pound of combustible burned.

W = Pounds of dry chimney gases per pound of combustible.

L_c = B.t.u. loss due to heat value of CO in chimney gases per pound of carbon.

L_{c1} = B.t.u. loss due to heat value of CO in chimney gases per pound of combustible burned from grate.

A = Pounds of air consumed in burning a weight of fuel containing a pound of carbon.

A_c = Theoretical pounds of air required to burn a pound of carbon.

A_h = Theoretical pounds of air required to burn a pound of hydrogen.

A_e = Pounds of air in excess of that theoretically required to burn a weight of fuel containing a pound of carbon.

$H_a = H - \frac{O}{8}$ in which

H = Pounds of hydrogen in a weight of fuel containing one pound of carbon.

O = Pounds of oxygen in a weight of fuel containing one pound of carbon.

That is H_a = that portion of hydrogen in the fuel which requires a supply of oxygen (or air) for its combustion. This calculation assumes that the oxygen content of the fuel is used for burning its equivalent of hydrogen. Then the remaining hydrogen (H_a) plus the carbon determines the amount of air that must be supplied for combustion through the furnace.

P = Volumetric percentage of CO_2 by Orsat analysis of chimney gases.
P_c = Volumetric percentage of CO by Orsat analysis of chimney gases.
S = Mean specific heat of dry chimney gases between the extremes of zero and perfect combustion.
T = Temperature Fahrenheit of gases escaping from boiler.
t = Temperature Fahrenheit of air in fireroom.

Deduction of Dry-Gas Chimney Loss in the Complete Combustion of One Pound of Pure Carbon to CO_2 with Varying Amounts of Air

$$L_d = W_d \times S\,(T - t).$$

For *perfect* combustion W_d is equal to the pound of carbon plus the air chemically required for its combustion. For the determination of the weight of this air we have the chemical equation for the combustion of carbon:—

$$C + O_2 = CO_2$$

Expressed in atomic weights of the elements we have:—

$$\left.\begin{array}{l} C + O_2 = CO_2 \\ 12 + 32 = 44 \end{array}\right\}$$ Hence for each weight or pound of carbon we must supply $\frac{32}{12} = 2.666$ weights or pounds of oxygen.

Now according to Sir William Ramsay, the composition of atmospheric air is 23.024 parts of oxygen and 75.539 parts of nitrogen and 1.437 of argon by weight. The nitrogen is inert and plays no part in combustion except as a diluent of the resulting gases, and argon as far as present knowledge goes is equally inert and may be treated as nitrogen in our combustion analysis. Hence calling the oxygen constituent of air 23 per cent by weight we must supply

$$\frac{2.666}{0.23} = 11.6 \text{ lb.}$$

of air to furnish sufficient oxygen for the perfect chemical combustion of a pound of carbon, and for this condition $W_d = 1 + 11.6 = 12.6$ pounds products of combustion per pound of carbon. The composition of these products will be

$$\begin{array}{l} 79\ N \\ 21\ CO_2 \end{array}$$

by volume and our Orsat apparatus would show 21 per cent CO_2.

This follows from Avagadro's Law which states that equal volumes of all gases under the same temperature and pressure contain the same number of molecules, and, conversely, when the number of molecules is un-

changed the volume is constant. Now when oxygen unites with carbon the number of molecules remain the same. Thus:— 1 molecule of carbon (C) + 1 molecule of oxygen (O_2) = 1 molecule of CO_2. Hence the volumetric relation of nitrogen to CO_2 or to a mixture of CO_2 + O in the combustion products remains constant and the same as its relation to O. It follows that with any excess of air from zero (perfect combustion) to infinity (no combustion) in the burning of pure carbon the volumetric percentages of CO_2 + O when no CO is present will always add up to 21. We have seen that:—

Perfect combustion of a pound of carbon requires 11.6 pounds of air, results in 12.6 pounds of products of combustion, and will show 21 per cent CO_2 by volumetric analysis. We must now relate the air supply and therefore W_d to the volumetric CO_2 of our CO_2 machine, by means of which together with a suitable thermometer in the flue gases we must be able to compute our chimney loss. Each molecule of CO_2 contains a molecule of oxygen and represents the satisfaction of chemical combustion requirements. Each molecule of free oxygen is of the same weight as the oxygen in the CO_2 molecule. Hence from our gas analysis we may readily compute the weight of air in excess of chemical

TABLE OF EXCESS AIR CORRESPONDING TO VARIOUS PERCENTAGES OF CO_2 IN FLUE GASES

For discussion see page 319

Volumetric Analysis of Combustion Products				*Air in Excess of Chemical Requirements*	*Lb. Air per Lb. C*	*Lb. Products of Combustion per Lb. C*
CO_2	O	CO	N[1]	*Per cent*	A	$1+A=W_a$
21	0	0	79	0	11.6	12.6
14	7	0	79	50	17.4	18.4
7	14	0	79	200	34.8	35.8
0	21	0	79	∞	∞	∞

[1] This includes argon as previously explained.

requirements, and knowing that one pound of carbon requires 11.6 pounds of air for perfect combustion, we may compile a table as exhibited on page 318 in which only a few results are given by way of illustration.

Next we must determine the specific heat of W_d. This depends upon its composition which varies from perfect to zero combustion as follows:—

Perfect combustion of one pound carbon results in

3.67 pounds CO_2	× specific heat 0.2170 =	. .0.796
8.93 pounds N	× specific heat 0.2438 =	. .2.180
12.60 total pounds products of combustion. .		2.976

$\div$ 12.6 = 0.236 specific heat at perfect combustion.
Zero combustion results in atmospheric air whose specific heat according to Regnault is 0.2375.

The mean specific heat of the flue gases between the extremes of combustion conditions is therefore $(0.236 + 0.2375) \div 2 = 0.237 = S$. If now we assume various flue gas temperatures and a fire-room temperature of say 80 degrees F. we shall have all the data required to construct an accurate set of curves which will show the true relation of chimney loss to volumetric percentages of CO_2 as found in the flue gas, when pure carbon is the fuel. An example is here quoted which will illustrate the method for

determination of the curves in Fig. 24 for pure carbon.

$$
\begin{aligned}
\text{Assume } T &= 400 \text{ degrees F.} \\
t &= 80 \text{ degrees F.} \\
CO_2 &= 7 \text{ per cent} \\
O &= 14 \text{ per cent} \\
CO &= 0 \text{ per cent}
\end{aligned}
$$

Substituting in the formula:—

$$
\begin{aligned}
L_d &= W_d \times S\,(T - t) \\
&= 35.8 \times 0.237\,(400 - 80) \\
&= 2{,}715 \text{ B.t.u., and the per cent loss based on the carbon burned will be}
\end{aligned}
$$

$$\frac{2{,}715}{14{,}600} = 18.6 \text{ per cent}$$

Chimney Loss in the Dry Gases from the Combustion of Actual Bituminous Coal. Deduction of a Set of Curves for a Selected Coal

The term combustible in this discussion is used according to the definition adopted by the American Society Mechanical Engineers, e. g., the coal *minus* ash and moisture. In a strict sense this is not a correct definition since the fuel contains oxygen which should not only *not* be rated as a combustible but *should* be rated as a *heat deductor*. That is,

the oxygen is likely to be in chemical union with the carbon before combustion, it may possibly be united with the hydrogen though not so likely, or it may be chemically combined with both hydrogen and carbon. (See discussion in Chapter XVI on fallacy of Dulong's formula.)

In any event, the oxygen content reduces the available heat value of a fuel by combining with and thus neutralizing a portion of its otherwise combustible elements. For this reason it is safe to rely for calorific determinations only upon actual oxygen bomb-calorimeter tests. This policy is maintained in the following deductions.

The question of air supply is a separate and distinct matter from that of chemical arrangement before combustion of the fuel constituents, but this however is directly influenced by the *amounts* of both hydrogen and oxygen in the fuel.

For convenience of computation we shall assume that *after* combustion, the oxygen constituent of the coal will be found united to its chemical equivalent of hydrogen. This oxygen, whether assumed to connect with the carbon or with the hydrogen in the final analysis, will in either event have exactly the same effect in reducing the air theoretically required to burn the fuel, and we shall have

to supply enough air to burn the carbon *plus* enough to burn the hydrogen after we have deducted that portion of the hydrogen content which will be satisfied by the oxygen already in the fuel.

This part of the hydrogen for which we must supply air is termed H_a in the following treatment, and it is evident that since the H_a will burn to H_2O which condenses in the Orsat machine, the original volume of oxygen supplied in air to the fuel will not appear as measurable in the Orsat; consequently the sum of $CO_2 + O$ (when no CO or hydrocarbons are present) will equal a percentage less than 21 depending upon the ratio of H_a to the carbon content of the fuel. The nitrogen of the air supplied to burn the H_a *does appear,* without, however, its normal complement of oxygen. This adds to the nitrogen ratio of the analyzable gas in the Orsat, thus reducing the percentage sum of $CO_2 + O$, zero CO and zero hydrocarbons being assumed as hypothesis in this treatment.

As an actual example illustrative of this action, in the products of combustion of natural gas which has a high hydrogen content I have found $CO_2 + O +$ (zero CO) equals as low as 14 per cent. If we burned pure hydrogen perfectly with no excess air our flue-gas

analysis would show 100 per cent nitrogen and zero oxygen. That is, the sum of $CO_2 + O$ would be zero, while for carbon under equally perfect conditions we would have 21 per cent CO_2 and 79 per cent N.[1]

With this introduction we may now proceed to a determination of the dry chimney loss as related to CO_2 and flue temperature resulting from the combustion of an actual sample of bituminous coal which has been selected as representative of good Pennsylvania fuel. In the following deductions it is assumed that all the hydrogen in the coal resolves to H_2O in the flue gas, and that neither CO nor any hydrocarbons are present. Sulphur is disregarded. "Combustible", it must be remembered, is coal *minus* ash and moisture.

In making boiler tests for commercial purposes where ordinary bituminous coal is used, these curves on Fig. 25 will give results far closer to accuracy than the usual pure-carbon curves. As elsewhere stated, the average error resulting from the use of the pure-carbon values as applied to this particular coal is equal in magnitude to that which would be occasioned by a mistake of 100 degrees in the observation of the flue temperature.

---

[1] N is used as the sum of nitrogen plus argon.

ANALYSIS OF THE COAL USED

From Pennsylvania, Allegheny County, Scott Haven, Ocean No. 2 Mine, Pittsburgh bed.

Mine sample collected by an inspector of the technologic branch of the Survey.

Analysis from Bulletin 22, published by the Department of the Interior, Bureau of Mines, Washington, D. C.

Analysis	*Sample as received*	*Dried at Temp. of 105° C.*	*Moisture and Ash Free*
	Per cent	*Per cent*	*Per cent*
Proximate:—Moisture	2.60	......	......
Volatile Matter	32.67	33.34	35.48
Fixed Carbon	59.41	61.00	64.52
Ash	5.32	5.46	......
	100.00	100.00	100.00

Analysis	*Sample as received*	*Dried at Temp. of 105° C.*	*Moisture and Ash Free*
Ultimate:—S	0.77	0.79	0.84
H	5.39	5.24	5.54
C	78.16	80.25	84.89
N	1.45	1.49	1.58
O	8.91	6.77	7.15
Ash	5.32	5.46	......
	100.00	100.00	100.00
Calorific Value B.t.u (By test in the Mahler bomb calorimeter.)	14,085 Per lb. as received	14,461 Per lb. of dry coal	15,297 Per lb. of combustible

From this analysis H = 5.54 per cent of the combustible, or for 1 lb. of combustible we have 0.0554 lb. of hydrogen. The oxygen on same basis = 0.0715 lb. Per weight of fuel containing 1 lb. of carbon we have $\frac{0.0554}{0.8489} = 0.0652$ lb. of hydrogen and $\frac{0.0715}{0.8489} = 0.0841$ lb. oxygen.

$$\text{Then } H_a = H - \frac{O}{8}$$

$$H_a = 0.0652 - \frac{0.0841}{8}$$

$$H_a = 0.055 \text{ lb.}$$

Now from the atomic weights of H and O we know that the burning of H to H_2O =

$\left\{\begin{array}{l} H_2 + O = H_2O \\ 2 + 16 = 18 \end{array}\right\}$ Hence 1 lb. of H requires 8 lb. of O, i.e. $\frac{8}{0.23} = 34.8$ lb. of air or 457 cu. ft.[1] for its combustion. The nitrogen in this amount of air will be $0.79 \times 457 = 361$ cu. ft. of N.

When 1 lb. of carbon burns to CO_2 we require, as before shown, 11.6 lb. of air or 152 cu. ft. From these data we may readily deduce that, for perfect combustion,

$$P = \frac{152 \times 21}{152 + 361\ H_a}$$

Reducing volumes to weights in this formula we have:

$$P = \frac{11.6 \times 21}{11.6 + 26.7\ H_a{}^{2}} \quad \text{for perfect}$$

combustion.

[1] Weight of 1 cu. ft. of air = 0.0761

[2] Weight of cu. ft. of N = 0.0738; 0.0738 × 361 = 26.7

The commercial accuracy of this second formula for P depends upon the very small error (amounting to a maximum of ⅓ of 1 per cent) which is introduced owing to the slight difference between the weight per cubic foot of air and of nitrogen. This error becomes only a fraction of the percentage difference between the above specific densities of the respective gases since the "H_a Nitrogen" forms only a small part of the denominator of the above expression for P, and this H_a nitrogen becomes relatively less as excess air beyond chemical requirements is added.

Substituting in this formula the H_a determined for our selected coal analysis we have:—

$$P = \frac{11.6 \times 21}{11.6 + (26.7 \times 0.055)}$$

$$P = 18.64$$

That is to say, the maximum CO_2 percentage obtainable in the burning of this coal will be *not* 21 per cent but *18.64 per cent.*

Now when excess air above chemical requirements is present the above formula becomes

$$P = \frac{11.6 \times 21}{11.6 + 26.7\ H_a + A_e}$$

Applying this to our special coal analysis in which $H_a = 0.055$, and selecting any

per cent of CO_2—say 10 per cent—as might be found on the Orsat we may solve for A_e as follows:—

$$A_e = \frac{243.6}{P} - 13.07, \text{ in which } P = 10$$

In which case the excess air (A_e) will be 11.29 pounds per weight of fuel containing a pound of carbon. The *percentage* excess air over that required will be this amount divided by it. The air theoretically required will be:—

$$A = A_c + (A_h \times H_a), \text{ in which}$$
$$A_c = 11.6 \text{ and } A_h = 34.8$$

Substituting these values:

$$A = 11.6 + (34.8 \times 0.055)$$

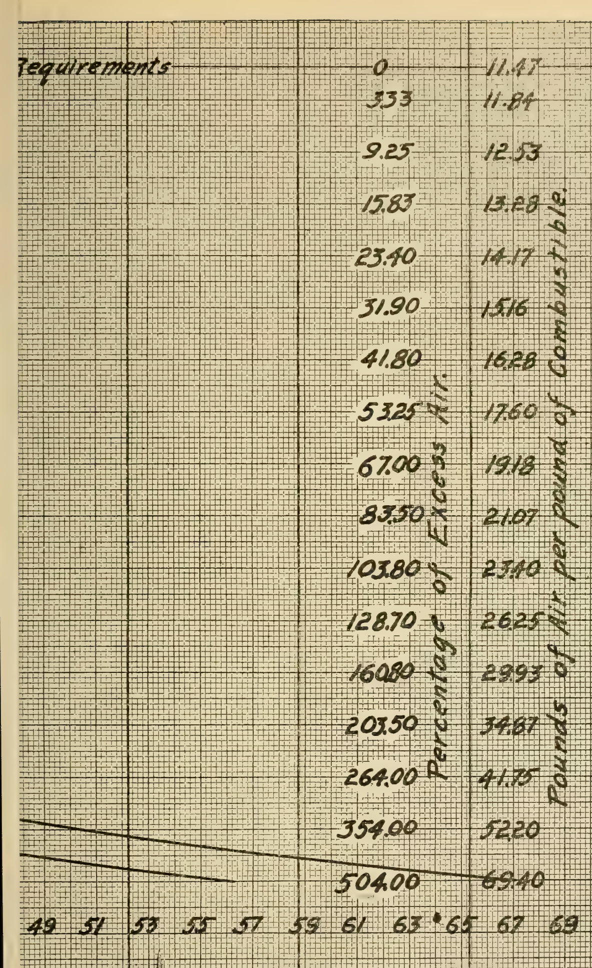
$A = 13.515$ lb. of air per weight of coal containing 1 lb. of carbon.

Hence when the CO_2 with this coal is 10 per cent, the excess air will be

$$\frac{A_e}{A} = \frac{11.29}{13.515} = 83.5$$

per cent above the amount theoretically required.

In the construction of the following coal curves P was computed in this manner for 17 different values from 3 per cent to the maximum of 18.64 per cent CO_2, together with all the related data given on Fig. 25. These may be compared with the pure-carbon curves of Fig. 24.



Requirements
0
3.33
9.25
15.83
23.40
31.90
41.80
53.25
67.00
83.50
103.80
128.70
160.80
203.50
264.00
354.00
504.00
11.47
11.84
12.53
13.28
14.17
15.16
16.28
17.60
19.18
21.07
23.40
26.25
29.93
34.87
41.75
52.20
69.40
Percentage of Excess Air.
Pounds of Air per pound of Combustible.
49 51 53 55 57 59 61 63 65 67 69
8000
9000
10,000.
le. (Heat Value 15,297 B.T.U. per pound.)

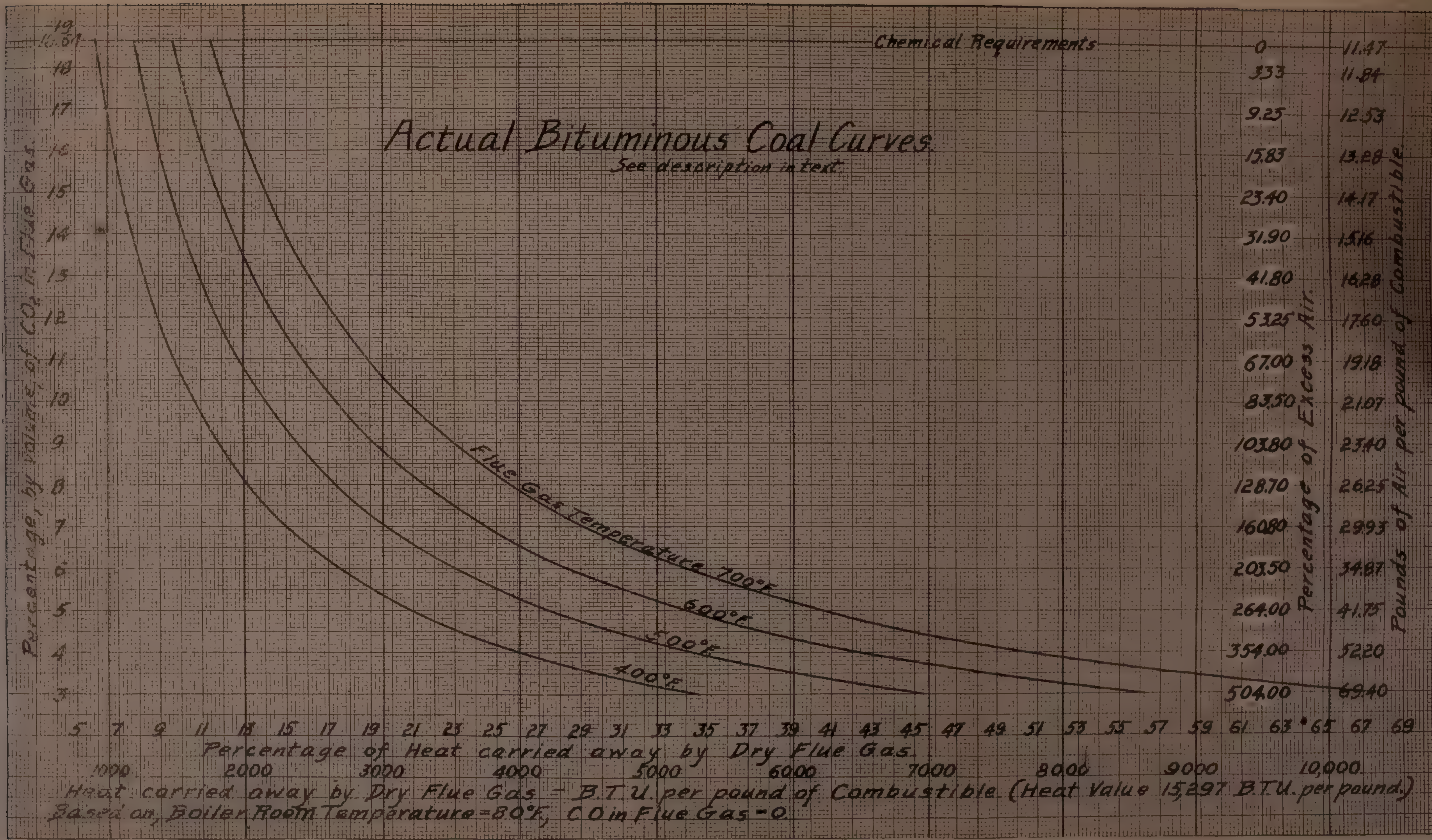


FIG. 25. ACTUAL BITUMINOUS-COAL CURVE

The heat carried away up the chimney for any percentage of CO_2 per weight of fuel containing one pound of carbon will be:—

$$L_d = W_d \times S\,(T - t)$$
$$W_d = A_c + 1 + A_e + 0.77\,(H_a \times A_h)$$

The last expression only of this formula demands explanation as follows: Only the nitrogen constituent of the air required to oxidize the H_a will be found in the dry portion of the chimney gases, the hydrogen and its equivalent oxygen reducing to water. This therefore amounts to 77 per cent by weight of the required air.

Substituting the values of our actual coal which prevail when $CO_2 = 10$ per cent as per previous deduction, we have:—

$$W_d = 11.6 + 1 + 11.29 + 0.77\,(0.055 \times 34.8)$$
W_d = 25.363 lb. of dry products of combustion from the burning of the special coal in question when the analysis of flue gases shows 10 per cent CO_2.

Before applying the formula for L_d we must determine S which is obtained thus:—

Perfect chemical combustion of this coal produces dry chimney gases from the W_d formula as below:—

12.6 lb. CO_2..........	$\begin{cases} 3.67 \text{ lb. } CO_2 \\ 8.93 \text{ lb. N} \end{cases}$		3.67 lb. $CO_2 \times 0.217$[1] = 0.796
	12.60		
$0.77\,(0.055 \times 34.8)$ =	1.47 lb. N	10.40 lb. N	$\times$ 0.244[2] = 2.539
		14.07 lb.	3.335

[1] Specific heat of CO_2.

[2] Specific heat of N.

$$\frac{3.335}{14.07} = 0.237 = \text{specific heat of dry gases when combustion is perfect.}$$

At the opposite extreme, i. e., zero combustion, the flue gas will be composed entirely of air whose specific heat is 0.2375.

Hence average specific heat of the flue gases between the two possible extremes will be 0.23725. As the error will be less than 0.10 per cent we may call $S = 0.237$.

To complete our example, assume the temperature of the fire room $t = 80$ degrees and that of the escaping flue gases 400 degrees F. and substitute the values corresponding to 10 per cent CO_2 in the formula:—

$$L_d = W_d \times S\,(T - t)$$

and we have

$$L_d = 25.363 \times 0.237\,(400 - 80)$$

$$L_d = 1{,}923 \text{ B.t.u. loss per weight of fuel containing 1 lb. of carbon.}$$

In this case the carbon constitutes 0.8489 of the combustible so that the loss per pound of combustible burned will be

$$L = 0.8489 \times L_d$$
$$= 1{,}633 \text{ B.t.u.}$$

We now have a complete analysis together with an actual example of the method employed to determine the values required for the construction of the curves on Fig. 25. It may now be seen by a comparison of the pure

carbon and the actual coal curves how great an error may occur when the former values are used for obtaining chimney losses in the burning of coal. Roughly speaking, there is a difference between the two sets of curves amounting to a percentage heat loss corresponding to about 100 degrees difference in flue temperature.

Determination of H_a in a Fuel from the Volumetric Analysis of the Flue Gases

For this solution samples of flue gas must be obtained which are free from hydrocarbons.

$$O_m = 21 - \left(CO_2 + O + \frac{CO^1}{2}\right) \quad \text{in which}$$

O_m = the volumetric percentage of oxygen absorbed by the H_a in the fuel.

The sum of $CO_2 + O + \frac{CO}{2}$ will equal the maximum volumetric percentage of CO_2 obtainable under perfect combustion with the fuel in question.

The formula:—

$$152 \times \frac{O_m}{CO_2 + O + \frac{CO}{2}}$$

---

[1] Oxygen when uniting with C to form CO doubles its number of molecules and hence its volume.

will then be the cubic feet of "measurable air" which furnished oxygen for the combustion of the H_a. Only the N of this air will appear in the Orsat burette which is therefore the quantity termed "measurable air". Now the burning of a pound of H results in 361 cubic feet of N (as before deduced), hence

$$H_a = \frac{152\ O_m}{\left(CO_2 + O + \frac{CO}{2}\right)361} \text{ which simplifies to}$$

$$\frac{O_m}{2.37\left(CO_2 + O + \frac{CO}{2}\right)}$$

To illustrate the use of this formula suppose our Orsat analysis should give

$$\left.\begin{array}{rl} CO_2 & = 8 \\ O & = 6 \\ CO & = 0 \\ \text{Hydrocarbons} & = 0 \end{array}\right\} \quad O_m = 21 - 14 = 7$$

Substituting these values we have:—

$$H_a = \frac{7}{2.37\ (8 + 6)} = 0.211 \text{ lb.}$$

The air required by such a fuel which coincides with tests on natural gas will be:—

$$\begin{array}{rl} 1 \text{ lb. C} \times 11.6 = & 11.60 \\ 0.211 \text{ lb. } H_a \times 34.8 = & 7.35 \\ \hline \end{array}$$

Total air required for chemical combustion of a weight of this fuel containing 1 lb. of carbon.................. 18.95 lb.

Now since we are able to obtain H_a from the analysis of a proper sample of flue gas, we may proceed to determine the *approximate* heating value of the fuel which produces these products of combustion as follows:—

Approximate B.t.u. of Combustible from Flue-gas Analysis

An amount of fuel containing 1 pound of carbon will have a heat value of:—

1 lb. C @ 14,600 B.t.u.	= 14,600 (B.t.u.)
H_a lb. @ 62,000 B.t.u.	= 62,000 H_a (B.t.u.)
Total heat	= 14,600 + 62,000 H_a

For example, take the case of the flue-gas analysis just quoted from which H_a was determined to be 0.211 pound.

Total heat = 14,600 + 0.211 × 62,000 = 27,682 B.t.u. in an amount of fuel containing 1 lb. of carbon.

Based on combustible we have approximately—

$$\frac{27{,}682}{1 + 0.211} = 22{,}830 \text{ B.t.u. per lb.}$$

A pound of natural gas from which such an analysis might be found would occupy about 21.9 cubic feet at atmospheric pressure.

$$\frac{22{,}830}{21.9} = 1{,}043 \text{ B.t.u. per cu. ft.}$$

which is a normally approximate result for a natural gas (giving the assumed flue-gas analysis) based on actual tests which I have made.

The B.t.u. of the gas thus determined is based on that part of the weight of the gas which is composed of C and H_a only, whereas it should, *if correct,* be based upon the *total* weight of the gas which includes unknown quantities of H, O, N, CO and CO_2 in various proportions.

With either coal or gas this flue-gas method develops another error common to both applications. *Since only the H_a portion of the H is multiplied by the heating value of hydrogen, and since the total C is multiplied by its heat value,* the result will be the same as if calculated by Dulong's formula (with sulphur disregarded) and will therefore include the same error. The cause of this error as explained in Chapter XVI is the incorrect assumption by Dulong that the oxygen in the fuel is pre-combined with the hydrogen, which has now been proved to be an incorrect hypothesis and leads to a heating value lower than would be shown by an actual test by bomb calorimeter.

An example for flue-gas determination of the heating value of coal will, however, be

of interest. Take the coal of the composition and heat value of our curves on Fig. 25.

B.t.u. per weight of fuel containing 1 lb. of carbon = 14,600 + $H_a \times 62{,}000$.

With zero hydrocarbons in the flue gas this coal would show:—

$$CO_2 + O + \frac{CO}{2} = 18.64 \quad \text{and}$$

$$O_m = 21 - 18.6 = 2.4.$$

$$\text{Then } H_a = \frac{2.4}{2.37 \times 18.6} = 0.055$$

We now have:—

B.t.u. in 1 lb. C	= 14,600
B.t.u. in 0.055 lb. H_a (62,000)	= 3,410
B.t.u. in fuel containing 1 lb. C	= 18,010

From the proximate analysis only we would not be able to relate this value to combustible, but since we have in this case the ultimate analysis we may do so thus:

0.8489 × 18,010 B.t.u. = 15,289 B.t.u. per lb. of combustible.

The flue-gas method of approximating fuel values is not to be recommended, by reason of the inaccuracies referred to as well as errors which occur in the Orsat determinations. The method is of interest principally on account of its scientific considerations.

From formulæ thus far deduced we are able to calculate within the limits of prac-

tical accuracy the amount of heat carried away in the "dry" chimney gases. We have now to consider the loss due to CO or half-burned carbon and also the heat carried away by the superheated steam resulting from moisture and hydrogen in the fuel.

Loss Due to CO in the Flue Gases

When there are no hydrocarbons present in the flue gases, the B.t.u. loss per weight of fuel containing 1 pound of carbon is:—

$$L_c = 10{,}150 \frac{P_c}{P + P_c}$$

in which

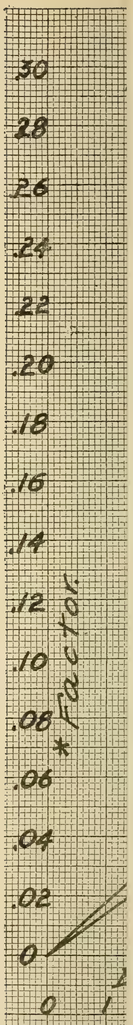
P_c = per cent of CO by volume from the Orsat.
P = per cent of CO_2 by volume from the Orsat.

This follows from the fact that a molecule or volume of CO contains the same weight of carbon as a molecule or volume of CO_2, and that a pound of carbon in CO when the combustion is completed to CO_2 develops 10,150 B.t.u.

Example

Flue-gas Analysis —	CO_2	=	15
	O	=	4
	CO	=	2
	H.C.	=	0

Then $L_c = 10{,}150 \dfrac{2}{15 + 2} = 1{,}194$ B.t.u. loss per weight of fuel containing 1 lb. of carbon.


.30
.28
.26
.24
.22
.20
.18
.16
.14
.12
.10
.08
.06
.04
.02
0
* Factor.
0
1
Percent

For facilitating this calculation the two curves of Fig. 26 have been plotted, one for pure carbon and one for the special analysis of coal which we have used throughout our discussion on combustion. To use either curve determine the factor

$$\frac{P_c}{P_c + P}$$

and apply on the ordinate to the upper curve for the coal and to the lower curve for carbon. The carbon curve is true for all coals when the result is corrected according to the percentage of carbon in the combustible, which must be had from an ultimate analysis of the fuel.

Wet Chimney Losses

The remaining chimney losses constitute the heat carried away by the moisture in the coal and by the moisture formed by the burning of its hydrogen content. These losses may preferably be based on combustible.

If X = pounds of water in the coal per pound of combustible, the moisture loss per pound of combustible will be:—

$$L_m = X\ [(212 - t) + 970.4 + 0.48\ (T - 212)]$$

The deduction of this formula is self-evident if it be considered that the moisture

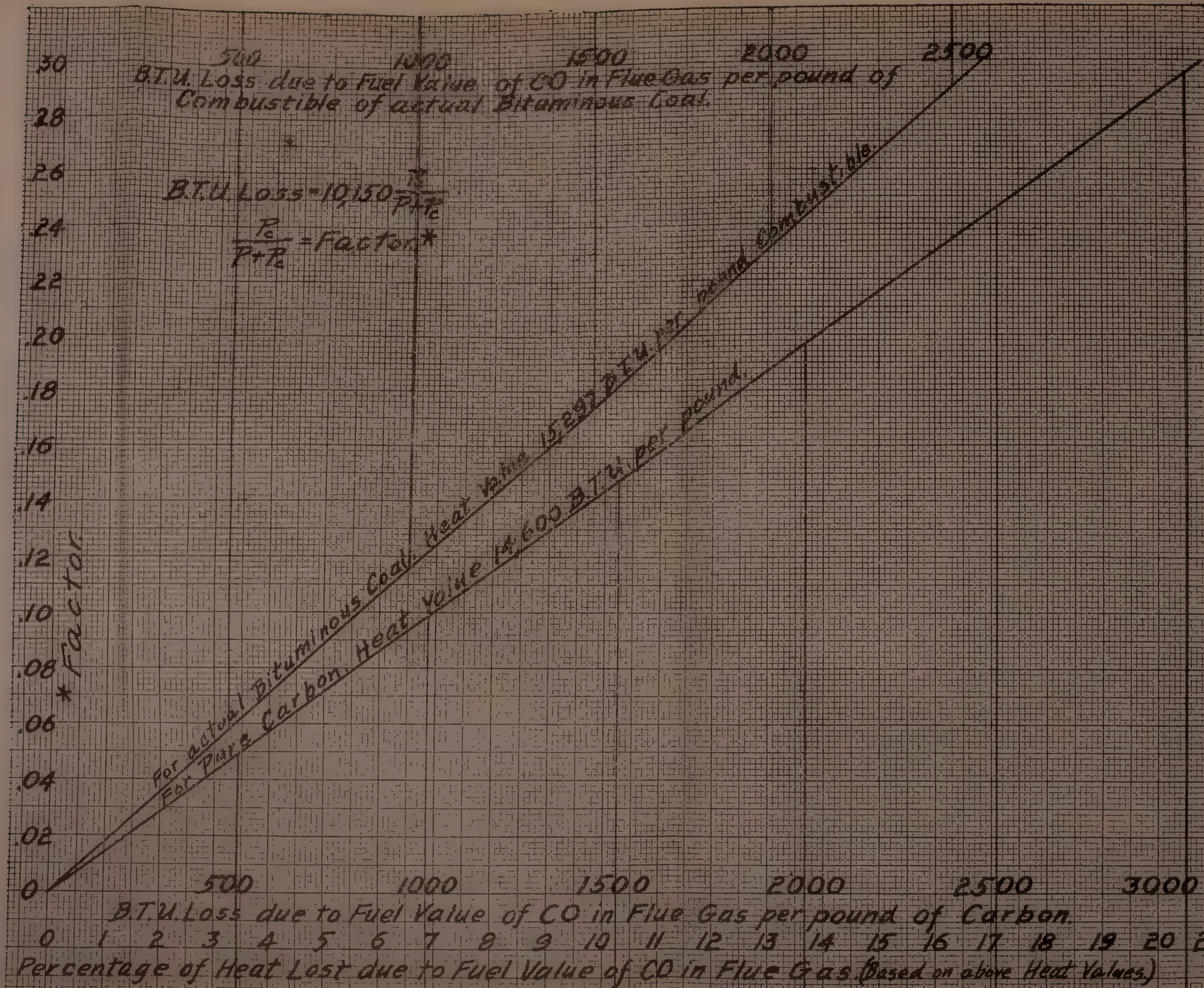


Fig. 26. Curves of Loss Due to CO in Flue Gases, for Pure Carbon and Actual Bituminous Coal

must first be raised from the fire-room temperature to 212 degrees (212 — t), then evaporated into steam from and at 212 degrees (970.4 = latent heat of steam), and finally this steam must be superheated to the temperature of the chimney gases, 0.48 being the specific heat of superheated steam at atmospheric pressure 0.48 (T — 212).

EXAMPLE:—Assume the coal to contain 1/10 lb. of moisture per lb. of combustible with t = 80 degrees and T = 400 degrees F. Then

$$L_m = 0.1\,[\,(212 - 80) + 970.4 + 0.48\,(400 - 212)\,]$$

L_m = 119 B.t.u. per lb. of combustible. If the combustible has a heating value of 15,000 B.t.u. per lb., then the loss due to moisture in the coal is $\frac{119}{15,000} = 0.794$ per cent of the heat of the combustible.

The moisture loss due to the burning of hydrogen is obtained by computing the amount of water which the hydrogen will form and then treating this amount the same as water in the foregoing formula. If H_x = pounds of hydrogen per pound of combustible, the pounds of water resulting from this hydrogen will be 9 H_x per pound of combustible, and the hydrogen moisture loss will be

$$L_h = 9\,H_x\;[(212 - t) + 970.4 + 0.48\,(T - 212)]$$

EXAMPLE:—In the coal analysis previously used H_x = 0.0554 lb.

$L_h = 9 \times 0.0554 [1193] = 595$ B.t.u. loss per lb. of combustible.

There is a further wet chimney loss due to the moisture in the air, but this is so insignificant a percentage of the total heat involved that its elimination from our heat balance will not affect the results within the limits of attainable accuracy.

In addition to the various chimney losses which we have now considered the heat balance of a boiler test involves the heat absorbed by the boiler [1] and a group of items which are usually classified as "unaccounted for", and which are generally found by deducting the sum of the other items from 100 per cent of the heat of the combustible.

Heat Balance of a Boiler and Furnace Test

1—Heat absorbed by boiler = heat utilized.

2—Heat in dry chimney gases.

3—Heat loss due to CO in chimney gases.

4—Heat loss due to moisture in fuel.

5—Heat loss due to moisture from hydrogen in fuel.

6—Heat loss due to unconsumed fuel, drop-

[1] Percentage of heat absorbed by the boiler per lb. of combustible = (evaporation from and at 212 degrees per lb. of combustible x 970.4) ÷ (heat value of a lb. of the combustible fed to the furnace).

ping through grate to ashpit and fuel removed through fire door when cleaning.

7—Heat contained in hot clinker and ash removed from furnace.

8—Heat radiated from boiler, furnace and setting.

9—Heat loss due to unconsumed hydrocarbons and hydrogen and to heating the moisture in the air.

The sum of the above values must constitute 100 per cent of the heat of the combustible fed to the furnace. Although items 6 to 9 inclusive are generally regarded as "unaccounted for" and obtained together by subtraction of the sum of the other items from 100 per cent, it is nevertheless possible to obtain a fair estimate of items 6 and 7 and sometimes of item 8. Item 6 may be obtained by taking a fair sample of the total ashes and clinker and making a calorific determination of its heating value. Then knowing their total weight, the loss due to the contained unconsumed fuel may be computed.

Item 7 is practically never recorded and credit for its separate inclusion and calculation is due to Mr. Albert A. Cary. If the temperature of this hot material as it leaves the furnace is known (and it can be fairly approximated) then its weight multiplied by

its specific heat and by the temperature difference involved will give the B.t.u. chargeable to this item.

Item 8—If the boiler is of an internally fired type, the radiation loss may be approximated by regarding the boiler as a steam radiator. The total heat that would be given off by the bare boiler may thus be computed, and if the thermal resistance of its lagging be known, a simple calculation in percentage will give a rough working figure of the actual radiation loss.

Chapter XIII

SURFACE COMBUSTION

FOR the future improvement of steam-boiler and furnace efficiency we may expect marked improvement by the employment of "surface" or "flameless" combustion, at least where gaseous fuel is available. In fact, the practice has already begun and several boilers in England and Germany are in operation under this system. This application of the invention has been announced only within the last three years.

As already discussed in Chapter XII on Combustion, the one greatest loss in the thermal operation of a boiler is the "chimney loss", which constitutes usually 20 to 30 per cent of the heat units in the fuel consumed. This loss is proportional to the weight of the hot gases escaping from the boiler.

Their weight is greatly increased in ordinary practice by the admission to the fur-

nace of more air than is actually required for the chemical combustion of the fuel constituents. This excess air must be introduced in order to insure contact of the oxygen with every particle of the fuel. If absolutely complete mixing of the air and the gases could be effected, this surplus air would not be required; but this is not possible to bring about even with the more scientific designs of common furnaces, with which it is usual to find one-third to one-half more air than is chemically needed. In poor and ordinary furnaces, indifferently operated, the air in excess of combustion requirements will range from 50 per cent to 400 per cent, with a consequently heavy chimney loss.

With surface combustion it is possible to furnish an air supply practically within chemical requirements. The weight of hot gases leaving the boiler is therefore a minimum for the fuel used and the attendant loss is proportionately reduced.

To Professor Bone of England and Professor Charles E. Lucke in the United States is attributed the recent development of this remarkable form of combustion. Professor Lucke has carried on an elaborate series of experiments leading to the application of surface combustion to various industrial and domestic purposes.

Professor Bone, among other devices, has developed a boiler which operates on the new principle, and the efficiencies combined with the capacities which he has obtained have exceeded the world's highest boiler-test results. He has shown an efficiency as high as 94.3 per cent, including the heat absorbed by a feed-water heater which forms a part of his device and is so arranged as to utilize some of the heat of the chimney gases. From available data it appears that this heater is responsible for conserving from 6 to 9 per cent of the heat of the fuel, and that the fan required for the draft may absorb 3 per cent of the steam developed by the boiler. To arrive at what may be considered *average net performance* of the Bone steam generator, it will be fair from various published tests to assume an everyday over-all efficiency of 93 per cent,—deduct 7 for the heater, which leaves 86 per cent, and then take 97 per cent of this to allow for the steam required to drive the fan. *This results in a net efficiency of 83.4 per cent,* and the important fact in connection with this result is that *it is obtained on a very small boiler, driven at a tremendous overload exceeding by many times all past performances.*

For instance, one boiler which was fired with oil (with arrangement for its gasifica-

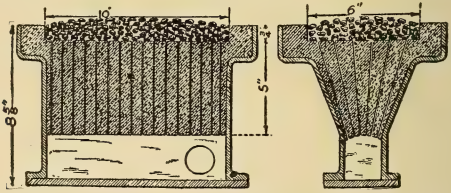
tion) was 5 feet in diameter and 12 feet long with five 9-inch boiler tubes. The total heating surface was 123.7 square feet, and it developed a total of 89 boiler horse power per hour, with an equivalent evaporation of 25 pounds of water per square foot of heating surface per hour. Other units with gas firing showed a rate of evaporation as high as 35 pounds per square foot per hour. Our present practice provides an evaporation of only from 3 to 6 pounds per square foot of surface, 3 being the more common figure. Hence the Bone boiler develops on an average about *seven* times the boiler horse power obtainable with other boilers of equal heating surface, and it does this at record-breaking efficiencies.

With the above data in mind we may now observe that the Bone boiler has developed almost six times the boiler horse power generated by the great Delray boilers per unit of heating surface, and this result was accomplished at an efficiency far higher than that of the Delray boilers even when operating at their most economical rating. Moreover, the latter are units of about 2,500 rated horse power each, whereas the Bone boilers were in units which developed from 50 horse power up to about 350 horse power.

Other tests on Professor Bone's boiler sub-

stantiate these results, and an evaporation as high as 35 pounds of water per square foot of heating surface has been obtained, together with an over-all efficiency of 93.8 per cent with coal gas as the fuel.

The operation of surface or flameless combustion consists in the pre-mixing of the fuel gas with almost the exact quantity of air prescribed by chemical equation, and then burning this explosive mixture in a bed of highly



Courtesy of *Steam*

FIG. 27. DR. LUCKE'S SYSTEM OF SURFACE COMBUSTION

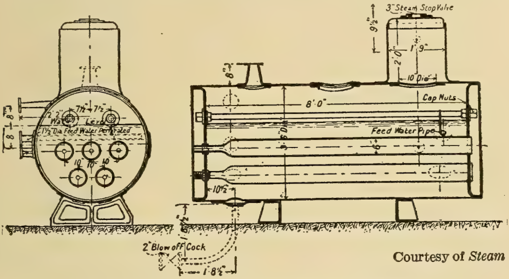
refractory granules which becomes incandescent at the zone of action.

As Dr. Lucke explains in his able treatise on the subject, it is necessary to introduce the gas mixture to the fuel bed at a velocity considerably higher than the rate of flame propagation. Otherwise the flame will back fire into the mixture chamber. Also if the gas velocity is not sufficiently reduced after its entrance into the combustion section, it will blow itself out by mechanically pushing

the burning gas out of the space provided for its consumption.

The former difficulty has been overcome by introducing the gas mixture through relatively long tubes of small diameter so that its velocity is always greater than that of flame propagation.

The second trouble, after careful experimenting, was obviated by adjustment of the



Courtesy of *Steam*

FIG. 28. SECTIONS OF BONE BOILER

shape of the combustion chamber, and the size and arrangement of the refractory granules, all of which constitute factors governing the velocity due both to mechanical and thermal expansion of the gases.

In Professor Bone's boiler here illustrated this feature is noted in the small supply tubes for the mixture, which in recent designs are surrounded by water in the boiler which

keeps their temperature low and thus further reduces the back-firing tendency. The process is known as flameless because the explosive mixture employed burns instantaneously (practically speaking) and an incandescent glow only is discernible. In ordinary combustion a flame is produced because

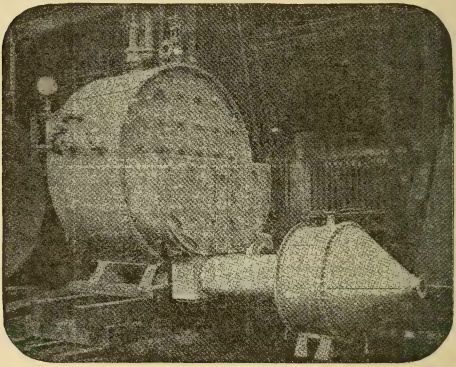


FIG. 29. FRONT VIEW OF BONE BOILER

time is required for the gases and oxygen to become mingled into intimate contact with each other, and actual burning occurs only upon such intimate contact. Consequently the process is progressive and slow and flame is observed.

In addition to the direct reduction of chimney loss by the exact proportioning of air

to the fuel made possible by this new process, there are other factors which contribute to the superior economies obtained. One of these, as may be judged from the illustration, is the practical elimination of radiation losses. The boiler is internally fired. Another consists in the extremely high temperature produced inside of the boiler tube where the combustion occurs in its bed of granules. This condition produces a greatly increased rate of heat transference from the combustion to the water in the boiler, since this rate is proportional to the difference in temperature between the hot gases and the absorbing body. But a still more important factor leading to the rapid heat-transfer obtained in the tests on the Bone boiler is the effect of *radiation* from the incandescent granules inside of each boiler tube. These granules attain the same temperature as the combustion gases and their "white heat" produces a transference more rapid than that which is produced by mere contact of gases equally hot but low in radiant heat.

Maximum temperatures are obtainable with this process owing to the absence of surplus air. The possible temperature of combustion may be computed from the heat units in the fuel together with the weight and

specific heat of the products of combustion. Thus:—

$$T = \frac{H}{W \times S}, \text{ in which}$$

H = B.t.u. per pound of the fuel.

W = Total weight of gases resulting from the combustion of a pound of the fuel.

S = Specific heat of the products of combustion.

T = Degrees F. rise in temperature due to the combustion.

Hence it is clear that by cutting the supply of air down to the theoretical minimum the products of combustion will also be a minimum and the attainable temperature a maximum.

As a matter of interest in this connection the records of the development of surface combustion show that a large amount of experimenting and research were required to discover the materials which in the form of granules would withstand the extreme temperatures produced.

The Bone boiler has been successfully operated on coal gas, coke-oven gas, and oil; plans are made for the consumption of other gases and there is no intrinsic reason why it should not be successful with all gas fuels. It is further stated that experimenting with solid fuel has been started.

It is difficult to predict just what effect flameless combustion will have upon our

present practice of coal-fired boilers. It was long ago proposed to convert coal into gas and then fire it under the boiler in a suitable furnace, with the hope that this method would produce decided economies over the direct firing of the coal into the boiler furnace. Of course gas is a nearly ideal form of fuel, and offers great advantages over solid fuel in that it lends itself to more intimate mixture with the air for combustion, consequently requires a less excess of air than coal, is far more readily regulated, and is not subject to the necessity and losses of furnace cleaning. Therefore its use tends toward higher efficiencies than those attendant upon the use of coal. And all these advantages count in its favor when a supply is available at prices sufficiently low to compete with coal.

But the conversion of coal into gas with its subsequent combustion develops an additional loss not connected with straight gas-burning. This loss attends the gas-producing process, which possesses an individual efficiency of its own. Therefore the gas derived from the coal contains less heat than the coal itself, so that the over-all efficiency of the producer-gas-fired boiler is at once handicapped by the gas-producing losses. And these losses have been of sufficient magnitude to

have prevented the attainment of as high efficiencies as are at present regularly obtained in well-designed stoker-fired steam-boiler equipments.

A producer for soft coal suitably designed for boiler firing may develop an efficiency of 90 per cent, providing it be so arranged as to make use of the sensible heat in the gases from the producer. That is to say, the gases which would leave the producer at about 1,200 degrees F. must not be cooled before their combustion under the boiler. It is evident, therefore, that even under best conditions an efficiency loss of 10 per cent must be deducted from the best result obtainable with straight gas-firing when the gas is made in a producer.

The best present practice in straight gas-fired boilers of moderate size indicates an efficiency of about 75 per cent, so that if we make the gas in a producer, our highest over-all efficiency becomes 0.90 × 75 per cent, or 67.5 per cent. It is apparent, therefore, why the producer-gas-fired boiler has not come into favor when its efficiency is compared to that of good stoker practice of 75 per cent efficiency which is readily obtainable with even small units.

Now let us investigate with reference to the possible over-all efficiency of a Bone sur-

face-combustion boiler operated on producer gas.

The net efficiency of the Bone boiler may be stated as 83.4 per cent, as per our previous analysis. Then with a producer efficiency of 90 per cent, as previously discussed, the probable maximum over-all efficiency of the Bone producer-gas-fired boiler would be $0.90 \times 83.4 = 75.1$ per cent. This would represent practically no saving over present stoker-fired boilers of *moderate size.* As compared to the large stoker-fired boilers like those of the Delray Edison plant, the Bone boiler with producer gas would operate at a loss as regards fuel consumption.

It would in truth save a large amount of space in the power plant owing to its very high steaming capacity per square foot of heating surface. Another marked advantage would be strongly evident in central-station practice where quickly changing and heavy peak loads are common; for each tube of the Bone boiler constitutes an individual furnace in itself. Consequently the variation in efficiency is but slight between the extremes of minimum and maximum output, for as the load increases or decreases an additional furnace or tube is fired or shut off completely, so that every such change has no effect on the efficiency of combustion and

causes only a very slight change in the efficiency of the boiler and furnace combined.

For purposes of comparison with common types of boilers the following standards of boiler and furnace efficiency may be used as representing the most refined practice with the respective fuels indicated.

For oil burning, the boiler tests at the plant of the Pacific Light and Power Company at Redondo, California (*Trans. A.S.M.E.* Vol. 33, Collins), gave combined boiler and furnace efficiencies ranging from 75.8 per cent to 83.3 per cent. The average efficiency of seven tests was 80.47 per cent, covering boiler ratings from 72.7 per cent up to 195.5 per cent of rated capacity. The highest efficiency (83.3 per cent) was obtained at 109.4 per cent of boiler rating; (builders' rating was 604 boiler horse power at about 10 square feet per horse power). It is to be noted that a deduction of 2.4 per cent for steam used in atomizing the oil must be made to arrive at the highest net efficiency obtained, which would be 83.3 per cent—2.4 per cent = 80.9 per cent, which may be taken to represent the best oil-burning practice.

For the most refined conditions of firing boilers with coal we may take the results obtained by Dr. Jacobus on the large stoker-fired boilers at the Delray station of the De-

troit Edison Company (*Trans. A.S.M.E.* Vol. 33, Jacobus). These boilers, the largest in operation at the present time, each contained 23,650 square feet of heating surface, giving a rated capacity of 2,365 boiler horse power at 10 square feet. The highest net efficiency obtained was (80.98 — 1.34[1]) 79.64 per cent in a 32-hour test. This was obtained at 98.6 per cent of boiler rating.

Isolated tests showing higher efficiencies than this have been obtained, some few claiming 83 or 84 per cent, but about 80 per cent combined boiler and furnace efficiency may fairly represent the highest results for coal burning under actual conditions of practice. Even in the above plant when the boilers were forced in accordance with good central-station practice, the efficiency dropped to between 75 and 76 per cent. The capacity of the boiler at these low efficiencies was about 200 per cent of rating, that is approximately 6 pounds of water were evaporated per square foot of heating surface per hour.

The efficiency of gas firing with small boilers may reach 75 per cent, and on large boilers like those of the Detroit Edison Company we might expect an efficiency somewhat higher than was obtained with coal. A net

[1] 1.34 per cent represents the steam used to operate the stoker and the draft.

efficiency of 83 per cent would be a good estimate for natural-gas firing under these largest of boiler units with suitable conditions.

The above comparisons are based upon the ability to utilize the sensible heat in the gases from a producer, and if this is accomplished the Bone boiler will be able to show marked economies over the average run of small size hand-fired boilers, on a basis of competition with automatic stokers, and it would certainly have the advantage of greatly increased capacity.

Possibly a coal furnace may be contrived to operate in conjunction with a surface-combustion system. Owing to a probable reduction of heat losses as compared to a producer, this field of endeavor should prove attractive to inventors. As before stated, experiments have been initiated looking toward the application of solid fuel to the Bone boiler. If success can be attained without serious counter losses the greatest field in the world will be open to the commercialization of this remarkable method of combustion.

The enormous evaporative capacity of the Bone boiler endows it with one great qualification for central-station service, and this fact will operate to increase the efforts of designers toward the direct and efficient application of coal for this special service.

Chapter XIV

NATURAL GAS AS A BOILER FUEL

IN those manufacturing regions where natural gas is available at low prices, it is also usual to find cheap coals of good quality. It is therefore necessary to decide which fuel will give the cheapest and most efficient service, and the problem is further complicated when the heating of the factory is considered in relation to the possible employment of gas engines. In every case arising for decision, the local operating conditions constitute one of the governing factors, and the price, heating value and reliability of supply of the respective fuels are of equal importance in their effect upon the final solution. The cost of attendance, initial investment, and other financial considerations, have their usual bearing upon the ultimate commercial efficiency; but our present object is to set forth the basic information more directly connected with the fuel itself, since it is evident that the

WEIGHT AND CALORIFIC VALUE OF VARIOUS GASES AT 32 DEGREES, FAHRENHEIT, AND ATMOSPHERIC PRESSURE WITH THEORETICAL AMOUNT OF AIR REQUIRED FOR COMBUSTION

Reproduced, by permission, from Babcock and Wilcox's "Steam"

Gas	*Symbol*	*Cubic Feet of Gas per Pound*	*B.t.u. per Pound*	*B.t.u. per Cubic Foot*
Hydrogen	H	177.90	62,000	349
Carbon monoxide	CO	12.81	4,369	341
Methane	CH_4	22.37	23,550	1,053
Acetylene	C_2H_2	13.79	21,465	1,556
Olefiant gas	C_2H_4	12.80	21,440	1,675
Ethane	C_2H_6	11.94	22,230	1,862

other considerations differ widely in accordance with the local conditions of any given case.

The composition of natural gas varies considerably, but by way of illustration the following analysis may be quoted:—

VOLUMETRIC ANALYSIS OF NATURAL GAS

CO	*H*	*CH_4*	*C_2H_4*	*CO_2*	*N*	*O*
0.50	2.18	92.6	0.31	0.26	3.61	0.34

An approximate estimate of the heating value of a gas may be obtained by adding together the individual heating powers of its combustible constituents. Thus for the above analysis we would have:—

Gas	*Per cent of Gas by Vol.*	*B.t.u. per cu. ft.*	*B.t.u. in Gas*
CO............	0.50	341	1.73
H.............	2.18	349	7.61
CH_4...........	92.60	1,053	975.07
C_2H_4..........	0.31	1,675	5.19

Total B.t.u. in gas per cu. ft. = 989.60

When ethane (C_2H_6) is present its heating value may be taken at 1,862 per cu. ft.

The table from Babcock and Wilcox's "Steam", reproduced on page 358, is useful in making estimates of this kind.

It is to be noted that the "high"[1] heating value of hydrogen is employed in the preceding data with its consequent effect upon the total heat value of the gas in question.

The calorific power of a gas is best determined by actual calorimeter test, in which the heat of combustion of a measured quantity is absorbed by water in a properly arranged jacket. When the temperature of the escaping products of combustion from such an apparatus is below the boiling point the heat measured will include the high value of the hydrogen content.

The following tests were made under my observation on a Sargent gas calorimeter. The fuel tested was natural gas at Lima, Ohio, supplied from wells in Ashland and Medina counties of that State:

Date of tests	June 13, 1914
Location	Lima Gas Works
Chemist in charge of tests	Mr. Rodney Lynch
Room temperature	79 degrees F.

Test No. 1

4.89 lb. water raised 19 degrees F. by burning 0.1 cu. ft. of the gas supplied at 70 degrees F. and measured at a pressure of 0.6 inch water gage. Barometer 29.6 inches.

B.t.u. per cu. ft. under these conditions, 966.

[1] See discussion of high heating value in Waste Fuel chapter.

Test No. 2

A few minutes later and under the same conditions a second test gave B.t.u. per cu. ft. as 1,004.

Other Tests

Other tests at the same place made a few days previous to the above gave calorific values per cu. ft. of 998 B.t.u. and 1,004 B.t.u. respectively.

For boiler purposes this gas sells according to a sliding scale of prices depending upon the rate of consumption, but the average price in practice at a boiler plant in Lima, Ohio, which I tested was about $0.125 per 1,000 cubic feet supplied and measured at a pressure varying from 8 to 16 ounces per square inch.

In Ashland, Ky., where the natural gas was rated by the gas company at 1,145 B.t.u. per cubic foot, it was sold for boiler firing at the rate of $0.10 per 1,000 cubic feet at meter pressure.

In Hambleton, West Va., the price was the same and the gas was rated at 1,123 B.t.u. per cubic foot.

The efficiency of natural gas as used for steam generation may be deduced from seven boiler tests which I made at the above named places.

Under actual conditions of firing a boiler horse-power hour was obtained from an av-

erage consumption of 40.4 cubic feet of gas at the metered pressure, the minimum being 37.3 cubic feet and the maximum 43.8 cubic feet.

With a good burner and a clean boiler there is no difficulty in obtaining combined boiler and furnace efficiencies from 70 to 74 per cent. Hence as compared to ordinary hand firing of soft coal, which gives an efficiency of 60 to 65 per cent, the gas shows to advantage in cost of evaporation, providing the same number of heat units may be purchased for a dollar. If opinion must be formed without any actual tests as to which fuel will produce the cheaper steam, it is best to base the comparison in case of hand firing on 63 per cent efficiency for the soft coal and 73 per cent for the gas. If on the other hand, a good stoker furnace were to be considered for the coal, it is safe to figure on equal efficiencies for both the coal and the gas in computing the fuel cost of evaporating 1,000 pounds of water. In practice these actually determined costs of evaporation with natural gas in the places mentioned ranged from \$0.0926 to \$0.1582.

The gas equivalent of coal in any locality depends upon the heat value of the coal and upon the heat value of a cubic foot of gas *at the metered pressure.*

The heating power of natural gas should be based on a unit volume at atmospheric pressure and a stated temperature usually 32 or 62 degrees F.

The most normal standard for general use is a temperature of 62 degrees and an atmospheric pressure of 14.7 pounds per square inch. The following formula is based on these values:—

$$Vo = \left(\frac{p \times 521}{14.7\,T}\right) V, \text{ which simplifies to}$$

$$Vo = \left(\frac{35.44\,p}{T}\right) V, \text{ in which}$$

Vo = the standardized volume in cu. ft.
p = absolute pressure at which the gas was metered in lb. per sq. in.
T = absolute temperature F. at which the gas was metered.
V = volume in cu. ft. of the gas as metered.

Calorific tests to determine the heating value of the gas are likely to be made at a temperature of about 62 degrees. This temperature standard is therefore more rational since the calorimeter results require less adjustment. The same argument holds good for boiler testing, since the gas will be metered at a temperature nearer 62 than 32. In any event, for commercial comparisons of the fuel values of gas and coal it is essential that the average pressure and tempera-

ture of the gas at the meter be known in order to arrive at a determination of the true heating value of the fuel as *purchased* at the meter.

The pressure factor is the more important, as may be seen from the following example in which the 62-degree standard is assumed. Thus if the gas is metered at one pound pressure, i. e., 15.7 pounds absolute, the heating value of a cubic foot will be nearly 7 per cent greater than the same quantity at atmospheric pressure, whereas a temperature rise even as great as 20 degrees will lower the heating value to the boilers by only 3.7 per cent.

If we desire to learn the gas equivalent of coal for any set of conditions the following example will serve as illustration of the method to be used.

The gas available has a heating value of 1,000 B.t.u. per cubic foot at atmospheric pressure, and a temperature of 62 degrees F. It is supplied at an average pressure of 1 pound per square inch at the meter and its average temperature is taken at 65 degrees. Its price is 10 cents per 1,000 cubic feet, as metered at the boiler room.

Applying the formula we have:—

$$Vo = \left(\frac{35.44 \times 15.7}{521}\right) V = 1.068 \text{ V}.$$

That is to say, a cubic foot as metered contains 1.068 times the weight or B.t.u. in a standard cubic foot. Hence a cubic foot as metered will contain $1.068 \times 1{,}000$ B.t.u. = 1,068 B.t.u. A thousand cubic feet as metered will therefore contain $1{,}000 \times 1{,}068 =$ 1,068,000 B.t.u. at a cost of 10 cents. *For $1.00 we may purchase 10,680,000 B.t.u. in natural gas.*

Now supposing the delivered price of coal in this locality is $2.00 per ton of 2,000 pounds having a calorific power of 13,500 B.t.u. after deducting 5 per cent of its B.t.u. per pound to allow for contained moisture. *Then $1.00 will purchase in coal 13,500,000 B.t.u. and in natural gas 10,680,000 B.t.u.* That is, with equal boiler and furnace efficiencies (assuming good stoker conditions in the case of coal) the coal would generate

$$\frac{13{,}500{,}000 - 10{,}680{,}000}{10{,}680{,}000}, \text{ or}$$

26.4 per cent more steam for the same expense for fuel. (Labor and investment comparisons must be made separately as conditions may demand.)

Without regard to prices, 1,000 cubic feet of natural gas under the above assumptions (which are characteristic of many gas regions) will have a heating value equal to that

of 79.1 pounds of 13,500 net B.t.u. coal. With natural gas and coal of the above specifications, coal at $2.53 per ton of 2,000 pounds would equal gas at $0.10 per 1,000 cubic feet.

When gas is compared to coal for boiler purposes no allowance need be made for different boiler and furnace efficiencies when the coal is stoker fired and the gas is used in a good furnace. When the comparison refers to ordinary hand-firing of soft coal, however, the calculation must allow for a lower efficiency of the coal. A common pair of figures would be 63 per cent efficiency for the coal and 73 per cent for the gas. Both fuels may burn at efficiencies either higher or lower than these, and the local conditions should be carefully investigated, preferably by actual test in order to insure a correct comparison.

Burners and furnaces for natural gas vary considerably in their design, and some of the home-made ones in my tests gave as good results as some of the patented ones.

The principal requirements for the efficient combustion of gas are as follows:

1. A thorough mixture of the gas with the required air. This is accomplished in three ways: (a) the burner proper may be so designed as to cause an excellent mixing action before the fuel enters the furnace; (b)

the mixing action may take place in the furnace itself, and it is usually augmented by baffles or checkerwork of firebrick which break up the currents of flow; (c) the cross-section of the furnace may be made large enough to compel a slow passage of the gas and air, thus providing time for their natural diffusion.

2. The maintenance of high temperature, which is accomplished by the impingement of the burning gas against hot brickwork, by its combustion within a firebrick furnace, and by retaining the gas in the furnace or combustion chamber until its combustion is practically completed.

3. Correct air supply. This also directly affects the second or temperature requirement. It may best be regulated both by actual testing of evaporative results and by making flue-gas analyses of the products of combustion. A rough and ready method consists in reducing the damper opening until smoke appears, and then opening the damper just wide enough to give a clear stack; but this method is not infallible, since it will not discover faults in the mixing action of the furnace as will tests by flue-gas analysis.

With a horizontal tubular boiler it is easier to furnish ample combustion space, because

of the long passage underneath, whereas with a water-tube boiler there is great danger that the gas and air may pass between the tubes before their combustion is effected. This bad result is guarded against by the interposition of checkerwork or baffles or by deepening the furnace.

A very light draft (that is, one of small pressure) gives the best results with natural gas, so that a low stack may be used successfully. One reason for this is that the resistance due to draft passage through a grate and a bed of fuel is eliminated, and this resistance constitutes roughly one-half of the entire drop in draft intensity through the complete boiler setting. There is usually sufficient pressure at the gas burners not only to impel the gas into the furnace, but also to cause an inspiratory action which tends to draw in to the furnace the air for combustion, so that, roughly speaking, the chimney need overcome only that part of the resistance which is offered by the passages through the boiler and flues. I have obtained 74 per cent efficiency on a horizontal tubular boiler operating at 90 per cent over its rating (at 12 square feet per horse power) with an average draft intensity of 0.2 inches water-gage measured between the damper and the boiler. It is best of course to provide more draft

than this in case of necessity such as might be caused by varying gas pressure, design of boiler and burners, pressure of gas at burners, and type of boiler.

The burners are usually inserted in the boiler front just above the coal grate, which

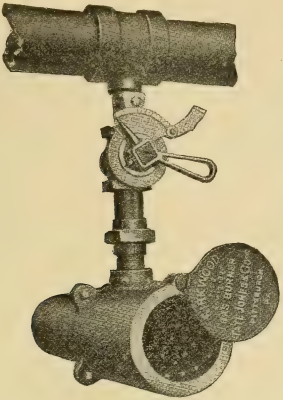
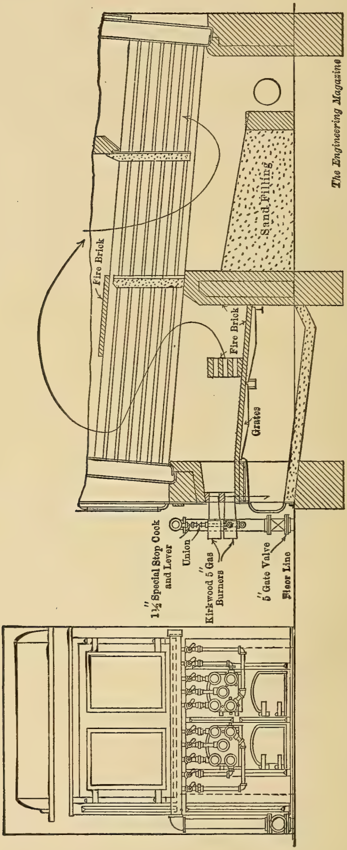


Fig. 30. The Kirkwood Burner

is bricked off, although with certain types of boilers like the vertical water-tube variety, the gas is fired through the front of Dutch-oven furnaces.

The burners themselves vary considerably in design. Two very good designs which I have tested are here illustrated. The Kirkwood system provides one burner for every



Front Elevation

Sectional Side Elevation

FIG. 31. KIRKWOOD GAS BURNERS APPLIED TO A WATER-TUBE BOILER

20 boiler horse power. The air enters through the large circular opening of each burner and the gas at a pressure of (preferably) about 8 ounces is fed to the annular

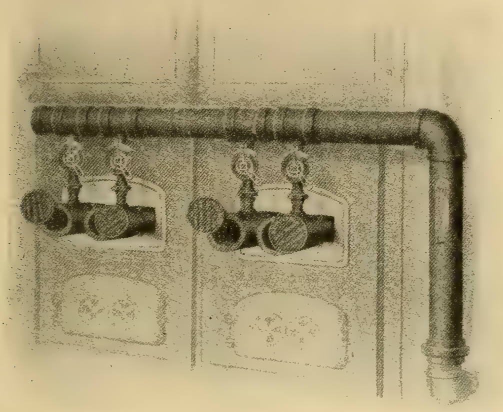


FIG. 32. TYPICAL SETTING OF KIRKWOOD BURNERS UNDER 100 HORSE-POWER BOILER

space which surrounds the air duct, and finds its way into the air tube through numerous communicating small holes which direct the gas into a corkscrew or whirling motion. This action aids the desired intimate mixing of the gas with the air. Both air and gas

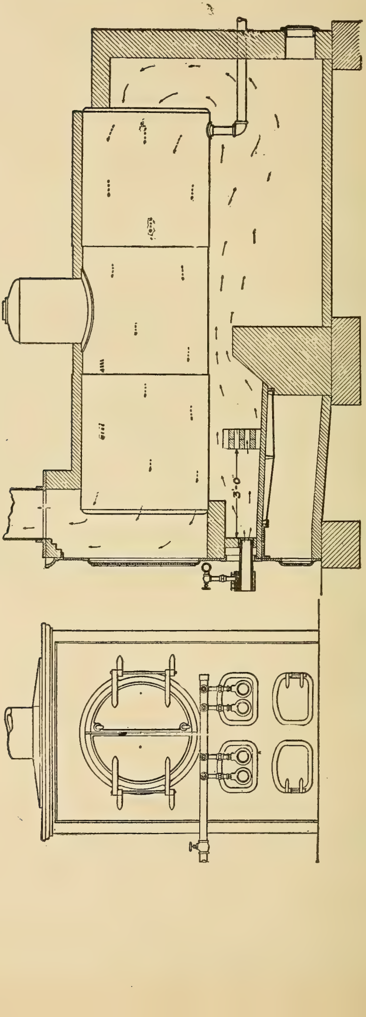


FIG. 33. GWYNN BURNERS APPLIED TO HORIZONTAL TUBULAR BOILER

are separately controlled, and in operation are so adjusted as to provide complete combustion at the mouth of each individual tube.

The most efficient regulation of the fire is obtained by completely shutting off or fully opening up one of these burner units. In this manner the ratio of air to fuel remains constant at all different ratings of the boiler.

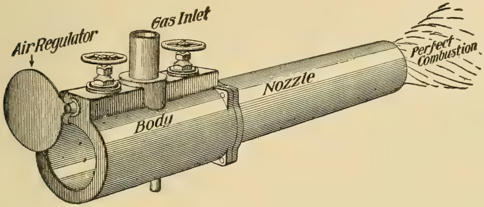


FIG. 34. GWYNN GAS BURNER

The Gwynn burner, as may be seen from the cuts, is constructed on the same general principles of operation just described.

Usual methods of furnace construction for gas burning are shown herewith as applied to horizontal tubular and water-tube boiler practise.

The bunsen type of burner is easily constructed from two pipes of different size; the larger is for the air and receives the gas at the centre of its opening. The inner end of this larger pipe or nipple leads just through

the boiler front. It is important that such burners be provided with individual covers for shutting off the air when the gas is stopped. Both good and bad results are obtained with these burners, depending principally upon the design and regulation of the furnace. One of my best and one of my worst tests were made on boilers supplied with home-made burners.

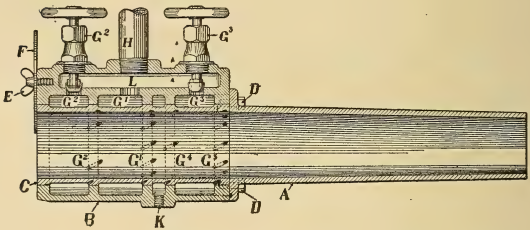


FIG. 35. SECTIONAL VIEW OF GWYNN GAS BURNER

Analyses of the products of combustion of natural gas provide a striking contrast with the results from coal. With pure carbon the volumetric percentages of CO_2 and O will add up to 21 when there is no CO present in the burner gases. With hard coal this total of 21 per cent is approximated. With bituminous coal the total will generally run from 18 to 20 when no CO is found. With the combustion of natural gas, of the compositions prevailing in my commercial tests on this

fuel, the total of $CO_2 + O$ when no CO was present has varied from about 13 to 16 per cent.

The reason for this dropping off of "total volume" from hard coal to soft, and again from soft coal to natural gas[1], is the greater proportion of hydrogen in the latter fuels, the gas containing the most. The burning of hydrogen results in the generation of superheated steam, thus $H_2 + O = H_2O$, and this steam condenses to water in the Orsat apparatus or before reaching it, so that its volume as a combustion product is not measurable in the analysis. Hence the greater the hydrogen constituent of a fuel, the less will be the sum of the gases determined by the usual Orsat type of gas machine.

From the above it may be properly concluded that with the same fractional surplus of air supplied to the combustion, the percentage of CO_2 from the burning of natural gas will be much lower than from the burning of coal, and this is the case in practice. In fact the best efficiencies have been obtained with natural gas when the average CO_2 was between 8 and 9 per cent, while in soft-coal burning with equal efficiencies 12 to 14 per cent of CO_2 would be expected.

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[1] See Chapter XII with actual example and calculation on natural gas.

RECORD OF BOILER TESTS AT ASHLAND, KENTUCKY

Below are recorded the records of five complete boiler tests on natural gas which I made for commercial purposes

	Test No. 1	*Test No. 2*
Water-heating surface	1,141 sq. ft.	1,500 sq. ft.
Type of boiler	Horizontal tubular	Atlas water-tube
Kind of burners	Home-made[1]	Home-made[1]
Location of plant	Ashland, Ky.	Ashland, Ky.
Total Quantities		
Date of trial	6/27/07	7/6/07
Duration of trial	7 hours	7 hours
Total weight of water fed to boiler	42,034 lb.	36,038 lb.
Factor of evaporation	1.0925	1.0904
Equivalent evaporation from and at 212 degrees F.	45,922 lb.	39,296 lb.

[1] Four of the home-made burners in these tests were attached to each boiler. Each burner consisted of a 1¼-inch gas supply pipe fitted on the end with a 3½-inch cap in which small holes were drilled, the cap being inserted into a 6-inch nipple which supplied the air.

Gas consumption	49,600 cu. ft.	51,200 cu. ft.
Gas pressure, inches of mercury	1.6	1.66
Gas consumption at atmospheric pressure	52,328 cu. ft.	54 032 cu. ft.
Hourly Quantities		
Gas consumed per hour (at pressure given above)	7,086 cu. ft.	7,314 cu. ft.
Gas consumed per hour at atmospheric pressure	7,475 cu. ft.	7,719 cu. ft.
Equivalent evaporation per hour from and at 212 degrees F.	6,560 lb.	5,614 lb.
Average Temperatures, Pressure, Etc.		
Steam pressure by gage	70 lb.	75 lb.
Temperature feed water entering boiler	155 degrees	158 degrees
Temperature of escaping gases from boiler	595 degrees	514 degrees
Force of draft between damper and boiler, inches of water	0.20	0.375
Horse power		
Boiler horse power developed	190	162.7
Rated horse power at 12 sq. ft. per horse power	95	150[1]
Percentage rated horse power developed	200	108

[1] This rating based on 10 square feet of heating surface per horse power.

RECORD OF BOILER TESTS AT ASHLAND, KENTUCKY—*Continued*

	Test No. 1	*Test No. 2*
Economic Results		
Water evaporated under actual conditions per cu. ft. of gas as fired	0.84746 lb.	0.70386 lb.
Water evaporated from and at 212 degrees F., per cu. ft. of gas as fired	0.9258 lb.	0.7675 lb.
Water evaporated from and at 212 degrees F., per cu. ft. gas at atmospheric pressure	0.8776 lb.	0.7273 lb.
Efficiency		
Calorific value of gas per cu. ft. (at atmospheric pressure)	1,145 B.t.u.	1,145 B.t.u.
Thermal efficiency of boiler and burner	74 per cent	61.4 per cent
Cost of Evaporation		
Cost of gas per 1,000 cu. ft. delivered in fire-room at above pressure	$0.10	$0.10
Cost of gas for evaporating 1,000 lb. of water from and at 212 degrees F.	$0.0926	$0.13
Gas required per boiler horse-power hour	37.3 cu. ft.	44.8 cu. ft.
Flue-gas Analysis		
Average CO_2	8.06	6.2
Average O	6.2	8.5
Average CO	0.2	0 to 0.4

FLUE-GAS SAMPLES

Test No. 1

Sample	Co_2	*O*	*CO*	*Flue Temp. Deg. F.*	*Draft in Uptake, In. of Water*
1	9.6	3.8	0	590	0.15
2	9.6	3.2	0.2	610	0.15
3	6.6	9.2	0	650	0.50
4	8.2	5.4	0	600	0.20
5	8.0	6.2	0.4	600	0.20
6	8.0	7.0	0.2	600	0.20
Test No. 2					
1	6.0	8.0	0	510	0.375
2	6.6	9.0	0.4	490	0.375
3	6.0	...	...	...	0.375
4	6.2	...	...	520	.....
5	7.4	7.6	0.8	...	0.25
6	7.8	7.4	0.4	...	0.25

RECORD OF THREE BOILER TESTS AT HAMBLETON, WEST VIRGINIA, ON SAME BOILER

	Test No. 3 [1]	*Test No. 4*	*Test No. 5*
Water-heating surface	1482 sq. ft.		
Type of boiler	Horizontal tubular		
Kind of burners	Gwynn		
Location of plant	Hambleton, West Virginia		
Total Quantities			
Date of trial	3/19/09	3/22/09	3/19/09
Duration of trial	7 hours	3 hours	3 hours
Total weight of water fed to boiler	33,194 lb.	15,694 lb.	14,818 lb.
Factor of evaporation	1.1506	1.1539	1.1537
Equivalent evaporation from and at 212 degrees F.	38,193 lb.	18,109 lb.	17,096 lb.
Gas consumption	43,752 cu. ft.	20,340 cu. ft.	18,952 cu. ft.
Gas pressure, inches of mercury	2.3	2.27	2.3
Gas consumption at atmospheric pressure	47,033 cu. ft.	21,845 cu. ft.	20,373 cu. ft.

[1] This test includes noon hour during which two of the four burners were shut off.

Hourly Quantities			
Gas consumed per hour (at pressure given above)	6,250 cu. ft.	6,780 cu. ft.	6,317 cu. ft.
Gas consumed per hour at atmospheric pressure	6,719 cu. ft.	7,282 cu. ft.	6,791 cu. ft.
Equivalent evaporation per hour from and at 212 degrees F.	5,456 lb.	6,036 lb.	5,699 lb.
Average Temperatures, Pressures, Etc.			
Steam pressure by gage	44 lb.	53 lb.	44 lb.
Temperature of feed water entering boiler, F	92 degrees	91 degrees	89 degrees
Temperature escaping gases from boiler, F	452 degrees	422 degrees	455 degrees
Force of draft between damper and boiler, inches of water	0.40	0.31	0.40
Temperature of air entering burners, F	66 degrees	....	....
Temperature of outside air, F	50 degrees	....	....
Horse Power			
Boiler horse power developed	158	175	165
Rated horse power at 12 sq. ft. per horse power	123	123.5	123.5
Percentage rated horse power developed	128	141.5	133.5

RECORD OF THREE BOILER TESTS AT HAMBLETON, WEST VIRGINIA, ON SAME BOILER—*Concluded*

	Test No. 3	*Test No. 4*	*Test No. 5*
Economic Results			
Water evaporated under actual conditions per cu. ft. of gas as fired	0.759 lb.	0.773 lb.	0.782 lb.
Water evaporated from and at 212 degrees F. per cu. ft. of gas as fired	0.873 lb.	0.8903 lb.	0.9022 lb.
Water evaporated from and at 212 degrees F., per cu. ft. gas at atmospheric pressure	0.812 lb.	0.829 lb.	0.839 lb.
Efficiency			
Calorific value of gas per cu. ft. (atmospheric pressure)	1,123 B.t.u.	1,123 B.t.u.	1,123 B.t.u.
Thermal efficiency of boiler and burner	69.8 per cent	71.3 per cent	72.1 per cent

Cost of Evaporation			
Cost of gas per 1,000 cu. ft. delivered in fire-room at above pressure..........	\$0.10	\$0.10	\$0.10
Cost of gas for evaporating 1,000 lb. of water from and at 212 degrees F......	\$0.1145	\$0.1123	\$0.1110
Gas required per boiler horse-power hour.	39.5 cu. ft.	38.8 cu. ft.	38.3 cu. ft.
Flue-gas Analysis			
Average CO_2........................	....	8.9	8.0
Average O..........................	....	5.0	7.2
Average CO.........................	....	0	0

The preceding records of five boiler tests which I made for commercial purposes are given to show results obtained under varying circumstances. These represent the operation of two horizontal tubular and one water-tube boiler, under different conditions as indicated by the observations on kind of burners, draft, boiler output, flue-gas analyses, etc.

Chapter XV

NATURAL GAS (*Continued*)

GAS ENGINE VERSUS STEAM ENGINE

IN the natural-gas regions manufacturers must determine whether this fuel will give the greater commercial efficiency when used directly in a gas engine or when applied to a boiler and steam engine.

The conditions of the problem will absolutely govern the results, but first let us compare the several over-all thermal efficiencies which are usually found in factory work, remembering that other things being equal the cost of power is inversely proportional to thermal efficiency.

Plant No. 1

Natural-gas-fired boilers. Steam engine, simple non-condensing. Exhaust steam wasted.

Boiler and furnace efficiency	=	73 per cent
Steam-engine efficiency	=	7 per cent
Over-all efficiency	=	5.11 per cent

Plant No. 2

Natural-gas-fired boilers. Steam engine, compound condensing or turbine. No exhaust-steam heating.

Plant No. 2—(*Continued*)

Boiler and furnace efficiency	= 73 per cent
Steam-engine efficiency	= 15 per cent
Over-all efficiency	= 10.95 per cent

Plant No. 3

Gas engine using natural gas. No by-product heat reclaimed.

Engine efficiency	= 20 per cent
Over-all efficiency	= 20 per cent

Plant No. 4

Natural-gas-fired boilers. Steam engine simple non-condensing. Exhaust steam efficiently utilized.

Boiler and furnace efficiency	= 73 per cent
Engine efficiency [1]	= 80 per cent
Over-all efficiency	= 58.4 per cent

It must be understood that the above assumed efficiencies will individually vary with excellence of equipment and operation, and consequently they must be regarded only as approximately typical of the designs of plant which they respectively represent. Bearing this precaution in mind, we may state that for the same gas consumption, a plant of the design of No. 4 will give nearly three times as much value as the most efficient of the others; that the gas engine of plant No. 3 will develop about four times as much energy as plant No. 1 and nearly twice the energy of plant No. 2 for the same amount of gas.

[1] See Utilization of Exhaust Steam, Chapters II and V.

But in actual practice the price charged for natural gas when used in a gas engine is much higher than when fired under a boiler. For example, in a recent case in Ohio the gas for boiler purposes cost \$0.125 and for gas engines about \$0.28 per 1,000 cubic feet. Hence with these prices, plant No. 1 would pay 0.125 ÷ 0.280, or nearly 45 per cent as much for a given amount of fuel, but this fuel would generate only 25½ per cent as much energy as it would develop if used in plant No. 3. Therefore the commercial efficiency of plant No. 1 would be far below that of the gas-engine plant in this instance.

In order to run the simple non-condensing engine of plant No. 1 at the same charge for gas as plant No. 3, the gas for boiler purposes would have to cost 5.11 ÷ 20, or only about 25½ per cent as much as when burned directly in a gas engine. That is to say, the prices of gas must be inversely proportional to the over-all thermal efficiencies of the different plants to make their commercial efficiencies equal.

The above classification shows at a glance very closely what may be expected from the use of natural gas in the different types of plants indicated where their respective efficiencies agree with those indicated.

From the above classification we may se-

lect the general type of plant which will give the highest commercial efficiency with natural gas when the two prices of gas are known, provided we alter the individual efficiencies given as necessity may demand.

There are of course plants of mixed design which may combine certain features of all four types. Also an existing plant of one of the types quoted may have, as previously indicated, an over-all thermal efficiency quite different from the efficiency designated for that plant in the foregoing classification. Consequently the certain way of arriving at best results for a given case is to base all computations on actual tests made upon both present and contemplated equipments. In the first place, such a method will have the advantage of discovering correctible losses in the existing conditions. Thus sometimes the efficiency of the old plant may be so improved at slight expense as to preclude the economy of the contemplated and more expensive change. For instance, if in a plant like No. 1 or No. 2 a sufficient use for exhaust steam can be made feasible, it would not pay to substitute a gas-engine plant like No. 3. An investigation would prove this and other matters connected with future economies.

The following example illustrates the test method in the solution of a problem.

Question: Will it pay to substitute a gas engine for a 100-kilowatt Corliss steam engine now running and wasting all of its exhaust steam, none of which can be used under existing conditions? The steam engine under test consumed an average of 28 pounds of steam per indicated horse-power hour, which average figure cannot be much improved under the imposed conditions of steam and load. The boiler supplying the engine shows under test an evaporation of 0.85 pounds of water per cubic foot of gas under actual commercial conditions. It is found to be developing an efficiency of 74 per cent which cannot be much improved for a 150 horse-power boiler under everyday working conditions.

Now from actual tests on factory gas engines of this capacity (see table of some of author's tests on factory gas engines) we know that we may expect a brake horse-power hour for a consumption of 12 cubic feet of gas under working conditions on the best design of engine. The prices of gas are 10 cents and 30 cents for boiler and gas-engine consumption respectively. The efficiency of the electric generator is 94 per cent.

We may then compute as shown in the calculation on the following page:—

COMPUTATION OF RELATIVE FUEL ECONOMY OF GAS ENGINE, AND STEAM ENGINE RUN BY GAS-FIRED BOILER

Steam Engine

Gas per indicated horse-power hour $\frac{(28.0)}{(0.85)}$	32.9 cu. ft.
Gas per kilowatt hour on switchboard (32.9×1.5)	49.4 cu. ft.
Cost of gas per kilowatt hour on switchboard — $\frac{49.4 \times 10}{1,000}$	\$0.00494

Gas Engine

Gas per brake horse-power hour	12 cu. ft.
Gas per kilowatt hour on switchboard (1.423×12)	17.1 cu. ft.
Cost of gas per kilowatt hour on switchboard — $\frac{17.1 \times 12}{1,000}$	\$0.0051

For further comment on these figures see page 393

AUTHOR'S TESTS ON GAS ENGINES

Five Engines of Four Different Makers

Manufacturers of Engine	*A. Company*	*A. Company*	*B. Company*	*C. Company*	*D. Company*
Gas consumption, natural, at 1 inch water pressure, cu. ft. per brake horse-power hour	14.2	13.1	12.5	11.3	10.5
Percentage of builders' rated horse power developed	90.3	95.8	55.0[1]	117.0	120.5
Number of cylinders	Single	Double	Single	Double	Double
Size of cylinders	$13\frac{1}{16}$ by 24	$14\frac{1}{2}$ by 24	$12\frac{3}{4}$ by 18	17 by 24	15 by 20
Method of ignition when tested	hot-tube	hot-tube and make and break	make and break	make and break	electric jump spark
Ignition efficiency at full load, per cent	100	100	100	100	100

[1] If run at its rated speed of 220 r.p.m. this engine would develop 22 per cent more power.

AUTHOR'S TESTS ON GAS ENGINES

Five Engines of Four Different Makers—*Continued*

Manufacturers of Engine	*A. Company*	*A. Company*	*B. Company*	*C. Company*	*D. Company*
Method of governing	hit and miss	hit and miss	hit and miss	hit and miss	throttle vac. gov.
Mechanical efficiency	74.5	87.2	76.4	80.8	88.3
Operation of air valve	suction	suction	mech.	mech.	mech.
Brake or actual horse power developed	27.1	95.8	22	176.4	90.4
Builders' rated horse power	30	100	40	150	75
Speed of engine in revolutions per minute	160	180	180	184	180

Under the conditions of prices and operation shown in the computation on page 390, the fuel for either the steam engine or the gas engine would cost almost the same amount, with a trifle of difference in favor of the steam engine.

Chapter XVI

THE ECONOMIC COMBUSTION OF WASTE FUELS

THE term "waste fuel" as used in this chapter means any combustible material which is not ordinarily included in the list of commercially marketable fuels.

Among the marketable fuels would be included bituminous and anthracite coals of the various grades and compositions; crude oil, fuel oil and the petroleum distillates, such as kerosene, gasoline and naphtha; the alcohols, grain (ethyl) and wood (methyl); some kinds and forms of wood; charcoal and coke; certain gases, such as natural gas, water gas, coal gas and producer gas; also peat and lignites.

As waste fuels would be classified the following: sawdust, shavings, scraps, edgings; tan bark, wood-extract chips; bagasse or spent sugar cane; anthracite coal passing through 3/32-inch mesh, known as culm; coke

braize, and city refuse. It will be noticed that these waste fuels (with the exception of culm and city refuse) are all by-products of manufacturing industries, and even culm might be so considered. City refuse is really a by-product of the various domestic industries inseparable from community living. So that the definition of waste fuel in a broad sense might be termed any combustible material resulting as a by-product of manufacture.

There are two distinct classes of waste fuels—"auto-combustibles," which maintain their own combustion after ignition, and combustibles which will not burn without the addition of heat from an outside source or without mixture with an auto-combustible. This second class may be termed for convenience "semi-combustibles." With the improvement of furnaces for the handling of waste fuels, certain materials which were previously considered only semi-combustible have joined the list of auto-combustibles.

The semi-combustibles resist self-ignition principally by reason of excessive moisture, ash, and refuse, and by physical structure tending either to resist the passage of the draft or to form large openings in the fuel bed which admit a great excess of air beyond that required for combustion, thus lowering

the furnace temperature and efficiency conditions.

A certain amount of heat must be given back to a substance in order to maintain its combustion. This is illustrated very nicely by removing suddenly the flame from a candle, which can be done either by a quick movement of air such as a puff of wind or by removing the air by smothering. The result is the same. The candle had been absorbing a part of the heat developed in the flame. As soon as this was no longer possible on account of the removing of the flame, combustion ceased, and to start it once more sufficient heat must be applied to secure the combustion temperature.

As a practical example, it is necessary, according to Stromeyer, to raise the temperature of lump coal to approximately 600 degrees Fahrenheit before ignition with the oxygen can take place. This figure multiplied by the specific heat of the coal would give the number of heat units required to be given by the fire to each pound of coal burned.

If what may be called the gross calorific value of a substance is actually less than the heat required to maintain its combustion, it is manifestly obvious that such a substance cannot be auto-combustible, and such sub-

stances, even when made combustible by application of heat from an outside source, will not give up any available heat for useful purposes since the amount produced is less than that actually required for maintaining their own combustion.

In the gasification of carbohydrates (i. e., fuels containing carbon, hydrogen and oxygen), which precedes combustion, a rather complicated chemical reaction takes place which either develops or absorbs heat. If the former, the reaction is known as exothermic; if the latter, endothermic. An exothermic reaction tends automatically to promote combustion, and the endothermic to retard. But since either condition is algebraically included in the calorific value of a substance, these interesting phenomena do not require separate consideration for the present purpose.

The amount of heat required to maintain combustion of a substance depends upon various factors. The first stage of combustion is gasification; therefore the heat required to gasify must be supplied in order to maintain combustion, and it is evident that the amount so required will depend upon the resistance to be overcome in order to gasify the substance. Moisture is a serious detriment to combustion, for the reason that it must be

gasified, requiring in addition to the sensible heat the application of 970 B.t.u. for each pound that must be removed, which heat must be obtained from the total heat value of the combustible.

Such substances as have been designated as semi-combustibles intrinsically possess a greater number of heat units than are theoretically required to maintain the combustion of said substances. This class of combustibles, among which are included certain kinds of refuse, wet chestnut chips, etc., may become actually auto-combustible, depending upon the efficiency of the conditions with which they are surrounded, such as hot surfaces, heated air supply, large combustion space, continuous feeding of the fuel and removal of ash, etc.

Thus some of these waste fuels will refuse to burn without the addition of coal or other outside fuel in *most* furnaces, and yet in a furnace which provides nearly perfect conditions for combustion, these same substances will burn and give off a large amount of available heat. As an illustration—wet spent chestnut chips until recent years would not burn unless they were supplied with additional heat from coal either by mixing or otherwise. Still another instance would be some kinds of city refuse, which when ordi-

narily treated refused to burn unless supplied with heat from a separate source. By the scientific design of furnaces and by careful homogeneous mixing of the various materials of this refuse, it is now burned without additional heat and at an efficiency high enough to make steam which can be used to operate the plant. These furnaces will be described in following pages.

Other fuels can be made more efficient by simply reducing the amount of moisture they contain without altering the design of furnace equipment. Special tests on moist fuels show close agreement between theory and practice in this respect.[1]

In conclusion of the above aspect of the problem, it can be said that the more nearly perfect are the means for burning the substance in question, the more nearly will semi-combustibles approach the state of auto-combustion.

Perhaps the most important of waste fuels in the United States has been spent tan bark. A rough estimate would indicate that this material generated a few years ago an amount of steam that would have otherwise required the yearly consumption of about 2,000,000 tons of high-grade coal. Yet this

[1] See author's paper, American Society of Mechanical Engineers, "Tan Bark as a Boiler Fuel."

valuable fuel was at one time considered a mere detriment and an expense to the leather industry. It was disposed of by dumping it into rivers, filling in waste ground, and by making roads with it, often necessitating the paying out of large sums for its disposition. This strikingly illustrates a case of how the improvement of a furnace converted a hitherto supposed incombustible into a valuable waste fuel of the auto-combustible class, and shows how an enormous waste was converted into an equally great economy.

An interesting example of the promotion of a previously considered semi-combustible to the list of auto-combustibles was the case of chestnut-wood extract chips. This fuel is the by-product resulting from the manufacture of a tanning agent largely and increasingly used in the manufacture of leather. Chestnut wood is first fed into chippers, then through disintegrators which reduce it to a finely divided state with particles about the size of No. 2 buckwheat coal. After being subjected for about a week to a hot-water treatment which extracts most of the available tannins, the spent chips are conveyed into the fireroom for burning. In this state they contain from 60 per cent to 65 per cent of moisture, and a dry pound shows about 8,400 B.t.u. That is about 1,100 B.t.u. less

than spent tan, which contains the same percentage of moisture. Ten per cent of this moisture can be removed by means of special presses for the purpose.

Ever since the institution of chestnut extract, the resulting chips were burned only by making with them a rich mixture of coal, or by maintaining a separate coal fire whose flames passed over the chips for the purpose of drying and igniting them.

In connection with this problem I designed a special type of furnace suited to the characteristics of this fuel. It then became possible to burn it without the addition, or mixture, of coal. In one of the foremost extract plants, this resulted in a great saving of coal and immediately advanced spent chestnut-extract chips to the ranks of auto-combustibles.

The matter of high efficiency in the utilization of by-product or waste fuels is just as important in establishments where such material furnishes only a part of their fuel supply as in plants that are operated entirely on commercial fuels, such as coal or gas. For in the first place, a highly efficient plant may be enabled to run entirely upon its by-product fuel material, whereas (as in my own experience) another plant in the same industry, of the same output capacity, but with

inefficient combustion equipment, may require additional fuel to the extent of many thousands of dollars a year. Furthermore, in a plant which depends partly on waste fuel supplemented by purchased fuel, any percentage saving that results from the improvement of the combustion system will produce an actual saving in the purchased fuel of this percentage multiplied by the fraction which represents the ratio of the total heating value of the combined fuel to the heating value of the purchased fuel.

For example, consider the case of a plant which consumes the equivalent of 100 tons of coal a day, of which heating value the purchased fuel constitutes one-third ($1/3$). Now if the efficiency of the furnace equipment is so improved as to produce the same steam or power for ten (10) per cent less fuel (or heat units), since the consumption of waste fuel still remains the same, the entire saving of fuel is effected from the purchased supply. In this case, therefore, in which a total saving of 10 per cent has been made, the expense of purchased fuel has been reduced not only 10 per cent, but $(3 \div 1) \times 10 = 30$ per cent. This simple arithmetical truth is but little if at all recognized by owners whose plants are operated on mixed fuels, or their combustion would receive far more attention.

The above considerations make the study of waste fuels and their efficient combustion most important.

Principles Involved

With waste fuels, in common with other fuels burned for boiler purposes, efficiency depends upon three conditions:

1. High temperature.
2. Correct amount of air supply, and
3. Complete mixture of this air with the fuel gases.

The great difference in the burning of waste fuels is not in the *requirements* for combustion, but in the *difficulty of meeting* these same requirements.

In the first place, temperature of combustion depends upon two factors:

a. The calorific value of the fuel, and

b. The amount of heat absorbed by its resulting products of combustion.

The temperature of combustion is directly proportional to the first and inversely proportional to the latter. The calorific value of waste fuel is generally comparatively low; hence the first difficulty in obtaining high temperature.

A larger percentage of moisture results in a greater weight of products of combustion,

and these products have a higher specific heat than flue gases, and, what is far more serious, involve also the latent heat of steam. Thus again the moist waste fuel presents further difficulty towards securing the first requirement for good combustion. In connection with this same requirement, the highest constant temperature can be maintained only when the fuel is introduced at the same rate at which combustion is taking place. With coal this is both possible and practicable, notably with certain forms of automatic stokers, and can even be approximated with skilful hand firing. But with waste fuels, special study and often much skill are required, owing to the great bulk and peculiar form of the material to be handled.

Again, with reference to the second requirement, that of *correct air supply,* waste fuel requires different treatment from coal. In the first place the weight of air theoretically required to oxidize or burn completely a pound of the fuel, depends upon its chemical composition. Thus carbon requires oxygen equal to 2 2/3 times its own weight, and hydrogen 8 times its own weight.

Beyond this theoretical air supply, an excess of from 50 to 250 per cent must be provided, according to the fuel in question and the design of the furnace. The latter should

be so constructed as to provide a thorough mixing of the air with the fuel gases at a high temperature, in order to reduce to a minimum the loss due to the great excess of air otherwise required. Still another difficulty has to be met in this connection. Certain waste fuels have a strong tendency to form blow holes, due either to clinker or lightness of the particles, in which latter case they are ripped from the grates and carried unburned through the flues and up the stack as waste. In either case, the blow holes lower the efficiency by admitting excessive amounts of cold air to the furnace, which reduce its temperature seriously. The opposite trouble occurs when the grates are too heavily loaded with a moist fuel that packs and checks the flow of the draft. Thus special construction and skill are required to overcome these difficulties, the occurrence of either one of which will utterly ruin the efficiency of the fire.

Another set of difficulties has to be overcome in connection with the third requirement of combustion, viz., *thorough mixing of the air supply* with the gases distilled from the fuel. This has already been touched upon. The requirement is rendered more or less difficult of satisfaction by two special factors, depending upon the nature of the fuel. First, the high percentage of volatile

gases evolved from many waste fuels; and secondly, the moisture content.

The tendency of the volatile portion is to flow quickly from the furnace, to become chilled below its ignition temperature by contact with boiler surfaces, and to escape unburned, thus constituting a serious loss of heat. One approved method for preventing this occurrence is so to construct the furnace and setting that predetermined currents will be produced which lengthen the travel of the gases, check their velocity, cause them to consume more time in contact with hot brick work in the presence of air, and finally produce a thorough mixture of the gases with the air for combustion. One of the simplest expedients for this purpose is the employment of very large and sometimes long combustion chambers. By merely increasing the cross-section of the passage, the velocity of the gases is reduced. This method is very simply and effectively illustrated by blowing through a ¼-inch pipe against the palm of the hand. The impact of the air is felt, denoting considerable velocity. This "velocity-pressure" indicates that the current of the air is rapid and that a particle or molecule passes through the small pipe very quickly. Now try the same experiment with a 1-inch pipe. The impact against the hand is light,

and this low velocity-pressure indicates that the air is moving slowly and taking a longer time to pass through the pipe having the larger section or draft passage. The pipe is the combustion chamber and the chimney is the same capacity in each case, that is to say, your own pair of lungs. The only difference in equipment is the cross-section of the combustion chamber.

By this method, therefore, the gases are compelled to remain a longer time in contact with hot surfaces, permitting more complete diffusion of the fuel gases with the oxygen, and thus combustion is promoted and improved. This feature combined with proper baffling to give a mechanical mixing action is productive of still better results. But all this must be accomplished before the fuel gases are allowed to come into intimate contact with the boiler cooling (generally called heating) surfaces. Other means are also employed to bring about this necessary mixing of the fuel gases with the air, among which may be cited special draft actions and furnaces constructed somewhat on the principle of the Argand burner.

The relation of moisture to the necessity of the mixing of the gases is interesting. The moisture is evaporated and becomes superheated steam gas. The molecules of this non-

combustible gas tend to form a separating medium between the molecules of oxygen and those of the combustible gases. Thus the element of moisture causes an added opposition to combustion and increases the necessity for, and value of, securing a thorough, homogeneous mixture by one means or another. This point was brought forward by Mr. Albert A. Cary, in his discussion before the American Society of Mechanical Engineers of my paper on "Tan Bark as a Boiler Fuel."

With this brief definition of the elements of the problem we shall next proceed to a description of the means actually employed in effective combustion of various waste fuels, beginning with sawdust and other waste wood products.

We shall now review the principal waste fuels, and show the means employed for their effective combustion or utilization. Taking them in the order first named, the characteristics and treatment of sawdust as fuel will be first considered.

Sawdust and Wood Waste

This material is of course available principally in the lumbering section. Together with the sawdust, the saw mills also produce

waste wood in the form of edgings, end cuts, shavings, and various forms of blocks and scraps.

The bulk of this material is so great that in many cases not only is the power and heating of the mill effected by it, but the whole town is furnished with electric lighting, and then in order to prevent the accumulation of waste wood a large destructor is kept constantly in operation, and the heat from the burning wood is wasted to the atmosphere. If there were a market for electric power within reasonable distance, this deplorable waste could be readily converted into a great economy. Even now plans are being considered, and at least one plant is in operation for producing wood alcohol and other chemicals with this material. Inventors have long worked on plans which might make possible its use as wood pulp in the paper industry.

Sawdust and wood waste are readily convertible into gas which may be used for power production in gas engines. Information regarding a specific case of this kind is given by Mr. Chas. E. Snypp in a paper contributed to the *Journal of the Association of Engineering Societies*, reviewed in the October, 1912, issue of *The Engineering Magazine*.

The burning of sawdust for boiler pur-

poses is best accomplished by means of a simply constructed "Dutch-oven" furnace. As a rule, only one feed hole in the roof of the furnace is used, for sawdust is such an excellent fuel that it will produce a hot fire even when roughly and carelessly treated.

Of course with a single feeding hole in the top of the Dutch oven, the sawdust forms a cone on the grate surface. This is about the least effective form in which any fuel may be arranged on a grate for burning. The draft penetrates the bed of fuel at the points of least resistance, that is, at the shallowest parts. The air, therefore, enters at an excessive rate around the edges of the cone, while the greatest portion of the grate is effectually air-tight so that this portion of the grate surface is "dead." This difficulty can be reduced or eliminated by different methods which will be later described. But as before stated, sawdust will respond with fairly good results to even this unskilful treatment.

In a mill where sawdust was plentiful, a very effective plant was inspected. A battery of six fire-tube boilers were each equipped with a Dutch-oven furnace, each having a single feed hole in its arch. An overhead conveyor brought in a continuous supply of sawdust. An iron chute from the con-

veyor to each furnace led the sawdust to a point about two feet directly above the feed hole of the furnace. The iron cover of this feed hole was left partly open, allowing the sawdust to flow into the furnace. By adjusting the sliding doors in the bottom of the conveyor, the flow of sawdust onto the grate could be made almost automatic. Of course a good deal of air entered the furnaces through the partly open feed holes in the tops of the arches, but the loss on this account would not be as great as might appear, for the reason that with a heavy pile of sawdust in the furnace a sufficient air supply would not enter through the grate and therefore the supply of air through the feed hole, although not admitted in a very scientific manner, was of real advantage. Intense combustion is actually seen to occur at the inside edges of the feed hole where the air meets the highly heated gases from the fuel.

Fig. 36 shows a typical sawdust furnace for horizontal tubular boilers.

Special problems in the combustion of wood waste often have to be met. One such case in my own experience is perhaps of interest, owing to the several rather opposing conditions that necessitated fulfillment all in a single furnace. It was desired to construct a burner which would handle with least labor

and greatest efficiency an amount of wood material containing sawdust, blocks, end-cuts, edgings, shavings, and slabs in varying proportions, and also it was further specified that the same furnace must be convertible into a good soft-coal burner at times when the wood fuel should not be available.

Fig. 37 is a cut of the furnace which was designed by me to meet these conditions and which is doing so in a satisfactory manner. Separate storage bins were provided for the sawdust and for the bulkier and miscellaneous materials. It was further arranged to feed the sawdust alone and separately through the top feed hole up to the full capacity of the furnace until this supply was depleted. Then by sealing the feed hole and charging by means of a "pusher" from a steel floor, level with the grates, the furnace handles, slabs, end-cuts, and the other forms of wood at good efficiency. The vertically sliding door being capable of quick opening and closing prevents excessive entrance of air during the feeding operation. When coal is fired, this is done through the same sliding door and the fuel is either spread or fired by the alternate or coking method.

As the grates are of a proper air spacing for fine sawdust, the correct regulating of the air supply for the other fuels used is accom-

plished by adjusting the admission of the required amount by means of the levers connecting the air valves on the front of the furnace. This air becomes pre-heated by the walls and combustion arch of the furnace, and is admitted to the fire at points surrounding the throat, thus producing somewhat the ef-

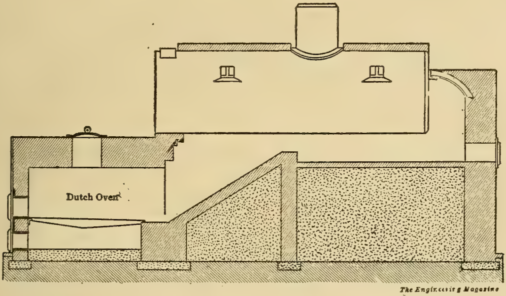

FIG. 36. SAWDUST FURNACE OF DUTCH-OVEN TYPE

From Steam Power-Plant Engineering, by G. F. Gebhardt. John Wiley & Sons

fect of an Argand burner. The large combustion space, baffle walls, and explosion doors complete the principal features of the equipment.

Although the air requirements and heat values of sawdust and coal differ greatly, either fuel will develop about the same number of heat units per square foot of grate surface per hour. It is largely this fact that

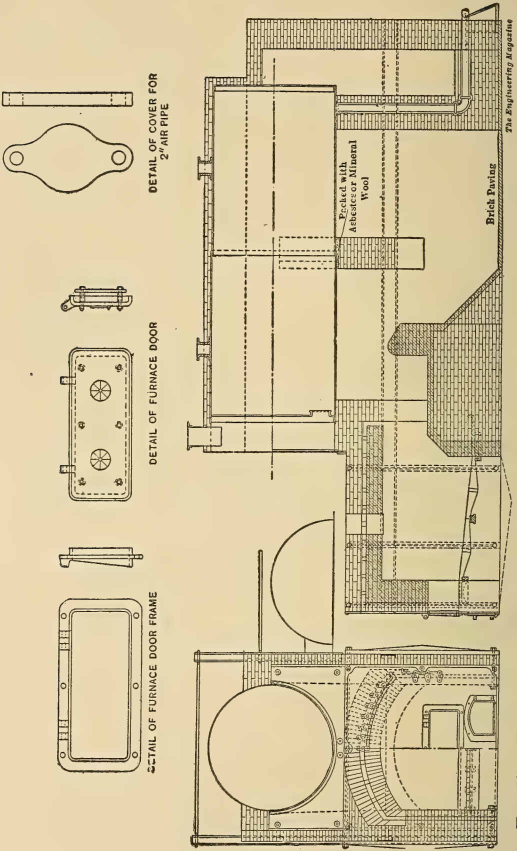

FIG. 37. FURNACE FOR BURNING WOOD WASTE, CONVERTIBLE INTO A GOOD SOFT-COAL BURNER

makes possible such an interchange of different fuels on the same grate and in the same furnace. The above furnace was reported by the owner to be consuming the calculated amount of wood-waste, and when this material is lacking the furnace burns soft coal without smoke. The high combustion is indicated by 12 per cent CO_2 as average in a month's run.

The heating value of wood depends upon the kind of wood in question and also upon its state of dryness.

"Hog feed" is the local term for saw-mill refuse which has been fed through a disintegrator or "hog." The various sizes and forms are thus reduced to a practically uniform size of chips, or rather shreds. In this condition it is greatly improved for convenience of handling, lends itself to higher efficiency in firing, and can be readily mixed and fed with sawdust to supplement the latter fuel in the same furnace. In the lumbering district of Wisconsin, I obtained samples of this material in the spring of the year, the product being the result of water-soaked timber from the mill pond. This material contained 52 per cent moisture, and 3.7 tons showed available heat value equivalent to 1 ton of 13,000 B.t.u. coal as tabulated on the following page:

Total heat per dry pound by bomb calorimeter test....................	8,600 B.t.u.
Moisture, per cent..................	52
Total heat in one moist pound.........	4,128 B.t.u.
To evaporate moisture to chimney temperature of 492 degrees F.......	650 B.t.u.
Heat available for boiler per wet pound as fired..........................	3,478 B.t.u.

(13,000 ÷ 3,478 = 3.74 tons hog feed = 1 ton coal.)

Figuring back on the dry wood basis, 100 pounds of dry hog feed containing 52 per cent moisture would produce 208 pounds of fuel at 3,478 available B.t.u. = 723,424 B.t.u., or one (1) lb. of dry wood produces an available heat in the fireroom of 7,234 B.t.u., so that tabulating the B.t.u. on four different bases we have:—

1. Heat per pound of dry wood........	8,600 B.t.u.
2. Available heat per pound of wet wood as fired.....................	3,478 B.t.u.
3. Available heat per pound of dry wood as fired.....................	7,234 B.t.u.
4. Total heat in a pound of the wet fuel.	4,128 B.t.u.

At this point I beg permission to deviate from the specific treatment of wood fuel in order to emphasize as clearly as possible a matter concerning which there is much confusion in recorded tests and in reference books in reporting the heat values of moist fuels.

The above table exemplifies four correct, though different, methods of reporting the

heating value of such fuels. Unless together with the value is also given a clear statement of how that value is obtained, the information is useless. In fact, it is worse than useless, as, four chances to one, it will cause a grievous and possibly fatal error in calculations intended to produce practical results. A glance at the values in this table proves that such an error may have a magnitude of 100 per cent above or below the truth.

There is indeed a fifth method of obtaining and reporting a heat value. In the above case the fuel was burned *after drying out* all the moisture. The moisture being separately determined, its loss by evaporation and temperature rise can be readily calculated for any set of actual furnace and boiler conditions. In the table on page 416 the temperature of the air entering the furnace was taken at 62 degrees and that of the escaping waste gases from the boiler at 492 degrees.

The fifth method is to burn the fuel in the bomb calorimeter *without removing its contained moisture.* This will give a value per pound of "wet fuel as fired" similar to, but higher than, value No. 2 in the table; because the moisture is re-condensed in the calorimeter and therefore gives up to the recording water that amount of heat which in the

actual case of burning under a boiler would pass off up the chimney as an actual loss.

In consequence of the above, it cannot be too strongly urged upon writers and engineers that they state exactly what is meant by the heat values they refer to when dealing with moist fuels. Along similar lines and with permission again asked, owing to the importance of truth in all things, the following point is brought to light.

If the calorific value of a fuel is determined by drying and then burning in a bomb calorimeter, the hydrogen contained in the substance is burned to steam gas (H_2O) which is condensed in the calorimeter. The result in this case gives what is known as the "high value," because the condensation of the steam produced by the burning of the hydrogen gives back to the calorimeter the latent heat of the steam thus produced, and this heat is measured as part of the heat value of the fuel. When, however, this same fuel is burned under a boiler from which the products of combustion escape into the atmosphere at a temperature higher than 212 degrees, the superheated steam formed by the burned hydrogen cannot condense and therefore carries with it out of the stack as a loss the latent heat of the steam (966 B.t.u.[1] per

[1] See footnote on page 421.

pound) as well as the heat that was taken from the fire to increase the temperature of this steam of combustion to the temperature of the escaping flue gases.

The high value of hydrogen as above described is 62,000 B.t.u. per pound. The low value is generally taken to mean the high value less the latent heat, without considering a final temperature higher than 212 degrees. This "low value" is 62,000 less the heat absorbed by the 9 pounds of steam (products of combustion of 1 pound of hydrogen). If the temperature of the hydrogen before burning is 39 degrees F., then the lost heat in the steam will be $9 \times ([212 - 39] + 966) = 10{,}251$, which deducted from 62,000 gives 51,749 B.t.u. as the "low" heating value of hydrogen. In some cases only the latent heat is deducted, which would make this figure 53,306 B.t.u.

Ordinarily, for commercial purposes, in calculating the available heat in a fuel, the loss due to burning hydrogen to steam gas is disregarded. When specified for some special purpose, however, in connection with a boiler test, it is calculated in the same way as moisture loss, each pound of hydrogen forming 9 pounds of steam or water.

The determination of the available heat in a moist fuel is as follows:

Let weight of fuel as fired............... = W lb.
Let percentage of moisture in fuel as fired... = M
Let heat value of dried fuel............... = H
Let temperature of fuel as fired.......... = t
Let temperature of flue gases............ = T

Total heat in one pound of the wet fuel is (W — MW) H = Total Heat (1).

Moisture Loss to be deducted from this total heat will be:

Loss, L = M ([212 — t] + 966[1] + 0.48 [T — 212]) (2).

Available heat per pound as fired (1) less (2) = (W — MW) H — L.

Example: Find the available heat per pound as fired for steam-making purposes, in a fuel containing 66 2/3 per cent moisture, and which shows when burned in a dry state in a bomb calorimeter a heat value of 9,600 B.t.u. per dry pound, assuming t = 62 degrees, T = 512 degrees.

Total heat in 1 pound of moist fuel = (9,600 — [0.66 2/3 × 9,600]) = 3,200 B.t.u. = (1).

Moisture Loss =

L = 0.666 ([212 — 62] + 966 + 0.48 [512 — 212]) = 840 B.t.u. = (2).

Available heat per pound as fired, 3,200 — 840 = 2,360 B.t.u. Answer.

Now in addition to the accurate bomb-calorimeter method of discovering the heating value of a fuel, upon which I base all my calculations, another but less accurate method

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[1] The work represented by this treatise on waste fuels was largely performed previous to the adoption of the Marks & Davis Steam Tables. Consequently the latent heat of steam here appears as 966 (accurately 965.7) B.t.u. rather than the modern value of 970.4 B.t.u. For all commercial purposes, however, the error introduced by this discrepancy being less than ½ of one per cent may be disregarded.

is employed. An ultimate analysis of the material is made by a competent chemist, which shows the percentage of each chemical element. Then as the heat value of each of these constituents is known, formulae have been devised for calculating with this information what may be the heat value of the substance as a whole. The most used formula for this purpose is that of Dulong, and is as follows:—

$$\text{B.t.u. per pound} = 14{,}600\ C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}000\ S,$$

in which C, H, O and S are respectively the percentages of carbon, hydrogen, oxygen and sulphur, each divided by 100.

It is my object to point out in the first place that this formula is not correct for fuels containing any considerable percentage of oxygen; and in the second place to show why this is true.

Take for practical example the ultimate analysis of tan bark, which is representative of woody fuels and is composed on the percentage basis as follows:— C = 51.8, H = 6.04, O = 40.74, Ash = 1.42. Apply Dulong's Formula and we have

$$146 + 51.8 + 620\left(6.04 - \frac{40.74}{8}\right) = 8{,}084 \text{ B.t.u.}$$

as the heat value thus derived.

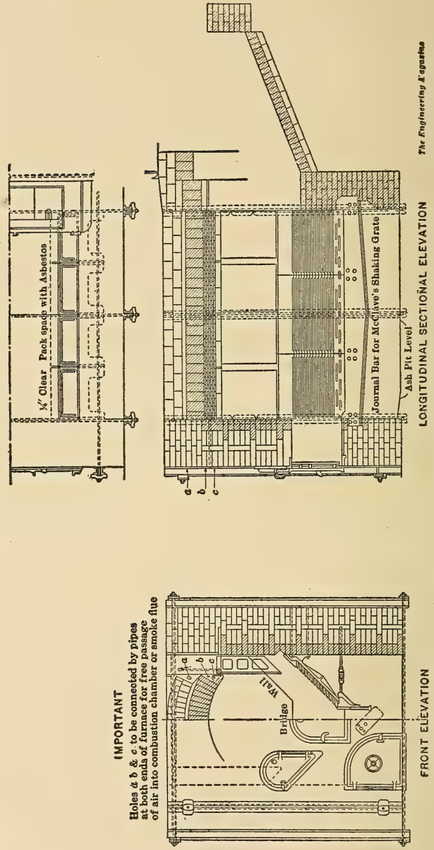


FIG. 38. SAWDUST BURNING FURNACE

Designed by the author to overcome inefficiency of draft distribution and inflow of unmixed air

Over forty actual bomb-calorimeter tests have shown, however, that the heat value is really 9,500 B.t.u., showing a discrepancy of the Dulong formula from the truth of nearly 15 per cent. So much for the actual fact. Now for the reason. Dulong's formula is based upon the erroneous assumption that *all* the oxygen in a compound exists in chemical combination with the hydrogen, forming H_2O, thus neutralizing the heat value of as many hydrogen atoms as are so combined.

During the writing of this paper Dr. Henry C. Sherman has written a most enlightening article on this subject under the title of "The Relation of Chemical Composition to Calorific Power in Wood, Peat and Similar Substances." Dr. Sherman's conclusions are based upon fifteen comparisons of calculated *versus* actual bomb-calorimeter values, and his summary which follows is the most authoritative statement on this subject at the present time. He states:—

It may be concluded

(1) That too much reliance should not be placed upon estimates of calorific power from ultimate chemical composition in fuels high in oxygen.

(2) That Dulong's formula, or any similar formula based on Welter's rule, of calculat-

ing the oxygen with the hydrogen, is likely to give results much below the truth.

(3) That the higher results obtained by calculating the oxygen of the sample as combining with the carbon, according to the suggestion of Walker, are much more nearly correct, and in most cases show a fair approximation to the values directly determined.

The foregoing discussion only emphasizes the necessity for accurate work and careful reasoning in dealing with waste fuels if accurate testing, high efficiency, and practical results are to be obtained.

To return to the actual handling of wood fuels, Fig. 36 shows an ordinary sawdust furnace of the Dutch-oven type such as is commonly found in the lumbering and saw-mill districts. It has the single top-feed hole and provides the characteristic cone-shaped bed of fuel on the grate. The inefficiency of draft distribution around the edges of this cone and the inflow of unmixed air through the feed hole have been discussed.

Fig. 38 shows a sawdust burner designed to overcome both of these difficulties and also to provide automatic feeding of the fuel at precisely the rate at which combustion takes place. The V-shaped grate receives two flat streams of sawdust from the lower open ends

of cast-iron or firebrick chutes of rectangular section which are kept filled with the fuel. As the fire consumes the sawdust, more is automatically supplied by gravity and a bed of fuel of unvarying thickness and efficiency is maintained. The thickness of the fuel can be changed at will to suit the draft by raising or lowering the feed chutes. Even draft distribution is obtained by the flat instead of the cone-shaped beds of sawdust, which greatly increases the rate of combustion and permits a smaller furnace being used. The grates are of the special type used in my stoker tan and chip furnace, and give a blow-pipe or concentrated draft action later described in connection with Fig. 42.

When operating on wet sawdust of 8,600 B.t.u. per dry pound, and containing 52 per cent moisture, the expected evaporation should be in the neighborhood of 2.5 pounds of water from and at 212 degrees per pound of sawdust, weighed as fired, with a combined thermal efficiency of boiler and furnace of 70 per cent of the available heat in a pound of the fuel as fired.

Tan Bark

It is believed that tan contains a larger percentage of moisture than any of the other

moist fuels. It is true, therefore, that its correct treatment as a fuel has valuable bearing on the correct handling of other moist fuels for high combustion efficiency.

The practical treatment of tan bark for high efficiency may be concisely outlined, first by a partial quotation from the summary of my paper on this subject read before the American Society of Mechanical Engineers at the annual meeting of 1909; and further by supplementary information with reference to illustrations of furnaces given herewith. The paper was the result of some forty tests on this fuel under actual conditions in various furnaces and settings.

Moisture:—In condition for firing, wet spent hemlock tan usually contains close to 65 per cent of moisture.

Available B.t.u.:—Bomb calorimeter tests on many samples of spent hemlock tan give an average value of about 9,500 B.t.u. per pound, samples being dried before burning. The average available heat per pound as fired, after subtracting moisture loss, is about 2,665 B.t.u.

Effect of Leaching on B.t.u.:—The heat value of spent hemlock tan is not affected by the degree of leaching, except inasmuch as the actual weight is affected.

Chemical Composition:—This has been

quoted in the previous discussion on Dulong's formula.

Improved Efficiency:—A considerable improvement in efficiency was produced by a specially designed furnace providing automatic feeding, large combustion space over the fuel and special draft admission.

Presses:—The use of presses for reducing the moisture before firing may constitute good economy if the amount of tan compared to the amount of coal is considerable, and providing the grate surface is properly reduced to meet the demands of more rapid combustion. The grate surface is sometimes reduced one-third. The gain in available heat is about $1\frac{1}{8}$ per cent for each per cent drop in moisture, and this drop rarely exceeds 7 per cent, i. e., a reduction of moisture from say 65 to 58 per cent.

Addition of Coal:—As a result of special comparative tests, the addition of about one pound of coal to six of pressed tan increased the combined furnace and boiler efficiency from 59.4 per cent to 63.4 per cent. The average CO_2 in the flue gases was raised from 10.9 to 13.8 per cent. The output of the boiler was increased from 92 per cent to 135.5 per cent of its rating. A further increase of efficiency would have resulted in a furnace

especially designed for burning a mixture of coal and tan.

Ample Combustion Space:—One of the most important factors in designing furnaces for tan-burning is that of ample combustion space. Low-arched furnaces are conducive to bad combustion.

Refractory Arch:—A refractory arch or similar combustion arrangement is essential. Tan in its usual condition cannot be burned in a common coal-burning setting without an arch separating the fire from the cooling surface of the boiler shell or tubes.

Furnace Temperature:—Excellent combustion of tan has given 1,500 degrees in the throat of the furnace.

Flue-Gas Analyses:—Basing comparison on flue-gas analyses, tan burns with a higher combustion than coal under equally favorable conditions. The large amount of moisture in tan produces comparatively low furnace temperature, even with good chemical combustion, and acts against an equally high combined efficiency of furnace and boiler.

The furnaces found throughout the country differed considerably in design and in results. Owing to the previous lack of testing no rationally fixed formulae existed, and consequently the furnaces were designed prin-

cipally on somebody's idea of what might give improved results.

All the furnaces had a Dutch-oven construction with different numbers of feed holes in the roof of the arch. Some were deep. Some were shallow. Occasionally a double-arched furnace was found. Combustion chambers were sometimes large, sometimes small; and systems of firing were found that suited the convenience and disposition of the

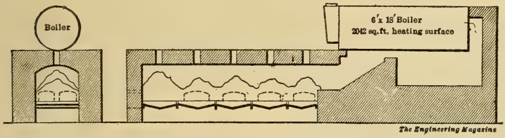


FIG. 39. EARLY TYPE OF FURNACE FOR BURNING TAN

firemen rather than the economy of the fuel and the design of the furnace.

Generally speaking the poorest results were found in low-arched furnaces and with heavy firing at long intervals. Much difficulty resulted from the cone-shaped beds of fuel heavily packed upon the grates with the draft ripping up the light edges into blow holes and cooling the furnace.

Ordinarily found types of tan furnaces are shown in Figs. 39 and 40. The first represents one of the earlier types, very long with

a single row of feed holes. This was known as the Hoyt furnace, and where there was a large excess of tan to be destroyed this furnace answered that purpose.

Fig. 40 is a more modern type with a double row of feed holes for more even distribution of fuel on the grate and shows a secondary air supply in the bridge-wall.

Fig. 42 shows my special design of automatic-stoker furnace which was successfully

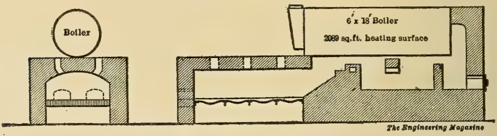


FIG. 40. LATER TYPE OF FURNACE FOR BURNING TAN

used in modern plants, and under test showed a combined efficiency of boiler and furnace of 71.1 per cent based on the available heat in the tan.

The essential features combining to secure these improved results, as will be noted from the cut, are: (1) large combustion space over the fuel bed; (2) automatic and continuous feeding of the fuel at the exact rate of combustion; (3) drying before receiving air supply by passing over dead plates at the upper edges of the grates; (4) discharge of ash by shaking grate at bottom of furnace;

and (5) special draft action as shown in Fig. 41. Owing to the horizontal air-spacing in the oppositely inclined grate surfaces, the draft currents arrive at a focus of combustion in the central zone of the furnace in a manner similar to the two flames of an acetylene gas

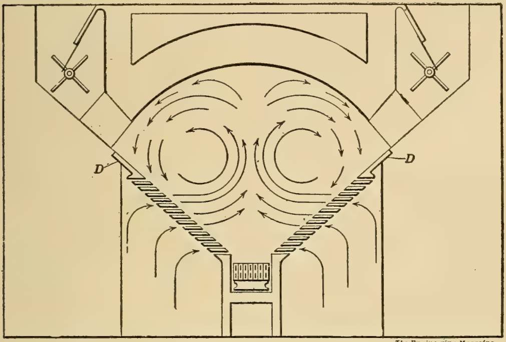


FIG. 41. SHOWING DRAFT ACTION OF FURNACE ILLUSTRATED IN FIG. 42

burner. By means of temperature differences and mechanical impulse, the flames or currents then react upon the dead plates, thus drying the fuel thereon; while the volatile gases that are distilled off are drawn downward into the focus of combustion where they meet the effective combination of high temperature and air supply and are thus consumed at full efficiency.

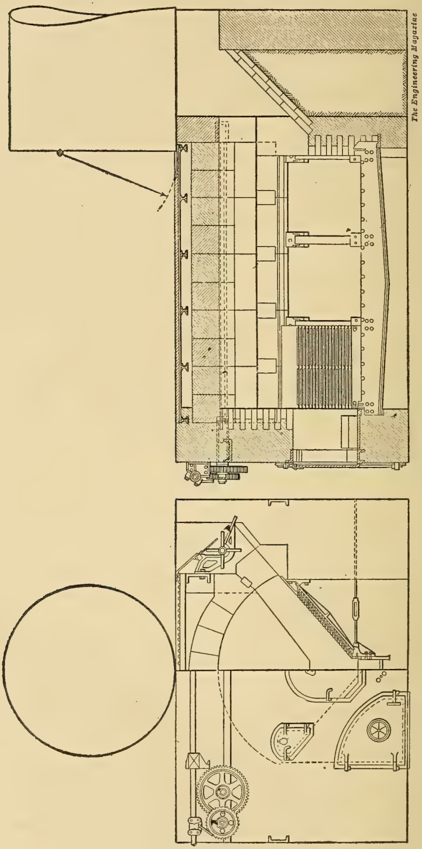


FIG. 42. AUTOMATIC-STOKER FURNACE FOR BURNING TAN BARK. FOR DRAFT ACTION SEE FIG. 41

Mixture Burning

Owing to shortage of tan bark and to increased use of steam in the modern tannery, it has been necessary to develop a mixture-burning furnace. Fig. 43 shows two of the author's recent installations designed to meet special requirements of this kind. Features to be noted are: (1) large combustion space; (2) a strong set of shaking grates as the clinker is bad with this kind of mixture; (3) heated auxiliary air supply; (4) special bridge wall for baffling; (5) small grate surface; and (6) provision for top feed through four holes and bottom front stoking by hand if desired. These furnaces operate with a heavy mixture of soft coal without smoke.

Chestnut-Extract Chips

Chestnut-extract chips are a more difficult fuel than tan, as they contain less heat and as much moisture. This fuel has been touched upon in another part of this paper and is one of those which have recently been changed from a semi-combustible to an auto-combustible. The author's tan furnace shown in Fig. 42 was successful in solving this particular problem.

This fuel was ordinarily handled in com-

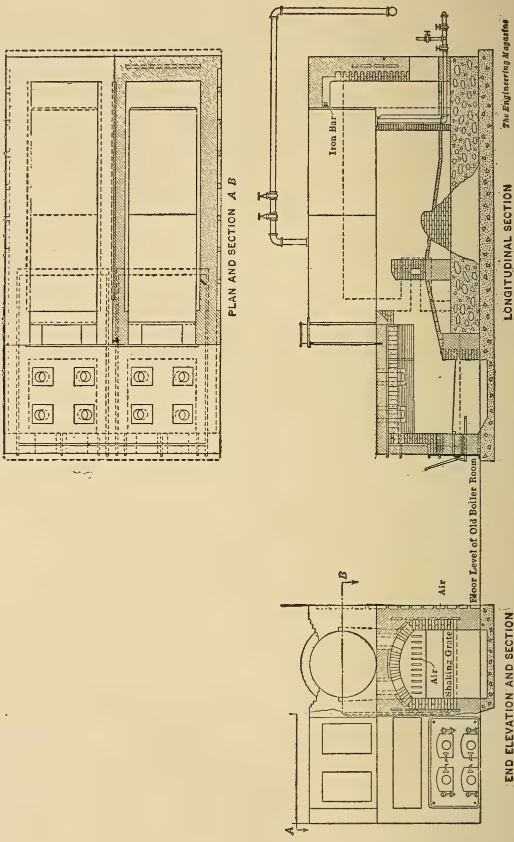


FIG. 43. FURNACE FOR BURNING MIXTURE OF TAN BARK AND COAL

mon tan furnaces, but would not burn without the application of coal. This extra fuel, of which large quantities were consumed, was in some cases mixed or "dumped" into the same furnace with the chips. In one of the large plants, the coal supply was stopped and when conditions became normal an evaporative test showed about *25 per cent* combined efficiency of boiler and furnace. Even *with* the coal, continuous and dense smoke was produced. The furnace had many feed holes but a very low arch.

In other cases the coal was burned on a separate grate in the rear of the chip furnace, and the heat thus developed ignited the chips and maintained a slow inefficient combustion.

In one plant eight of the automatic-stoker tan furnaces, before referred to, burned these chips without any coal, with a combustion efficiency which produced continuous readings of 12 to 14 per cent CO_2, developed no smoke, gave over full rating of boiler horse power, and showed an operating saving of many thousands of dollars a year over a duplicate plant using the old-time furnaces with auxiliary coal grates. These observations, from the commercial standpoint, emphasize the important relation of scientific furnace-design to industrial efficiency.

Licorice Chips

Extracted or spent licorice-root chips are similar in general composition to chestnut chips, and I inspected a furnace that was very successful in burning the former fuel at high efficiency. The construction was along the general lines of an Argand burner. The fuel was fed into a Dutch oven in a heavy bed wherein sufficient temperature was maintained to gasify the chips. The resultant gases flowed through the furnace throat into a cylindrical combustion chamber, in whose centre was a hollow checker-work of brick by means of which a preheated air supply entered and mixed with the fuel gases. The brick checker-work of the Argand burner together with the combustion chamber walls maintained a high temperature, thus promoting the ignition and combustion of the gases which were well mixed with the preheated air supply by means of the Argand feature.

Bagasse

Bagasse is the by-product fuel resulting from the manufacture of cane sugar. The juice from the ripe cane is extracted by crushing in powerful mills. The remaining fibrous material, known as bagasse, is conveyed to the boiler house where it is fired in

special furnaces and is made to produce all, or part of, the steam required to operate the plant. The fuel is in the form of strips from 3 to 8 inches long, but very much longer pieces may come through the mills.

In regard to the heat value of bagasse, Dr. Sherman made determinations on a Cuban sample imported by the author in an air-tight can (to retain the moisture). This sample showed:—B.t.u. per dry pound, 8,324; moisture, 57.9 per cent.

If this were burned under a boiler giving a flue temperature of 594 degrees Fahrenheit, the heat per pound of fuel, weighed as fired, available for steam purposes would be ([100 per cent — 57.9 per cent] 8,324 — 57.9 per cent ([212—62] + 970 + 0.48 [594—212]) = 2,750 B.t.u.

Mr. E. W. Kerr, who has made a study of bagasse burning in different styles of furnaces, is quoted from Bulletin 117 of the Louisiana Agricultural Experiment Station as follows:

"An equivalent evaporation of 2¼ pounds of steam from and at 212 degrees was obtained from 1 pound of wet bagasse of a net calorific value of 3,256 B.t.u. This net value is that calculated from the analysis by Dulong's [1] formula, minus the heat required to

[1] See author's discussion of Dulong's formula, pages 421–424.

evaporate the moisture and to heat the vapor to the temperature of the escaping flue gases, 594 degrees Fahrenheit. The approximate composition of bagasse of 75 per cent extraction is given as 51 per cent free moisture, and 28 per cent of water combined with 21 per cent of carbon in the fibre and sugar." "For the best results bagasse should be burned at a high rate of combustion at least 100 pounds per square foot of grate per hour."

For obtaining quick combustion mechanical draft is used in many plants, the draft being admitted to the fuel from under the grates and also through tuyeres in the furnace walls. This is illustrated in Fig. 44, kindly contributed by a test record by Dr. D. S. Jacobus, Advisory Engineer of the B. & W. Boiler Company. This furnace is shown in connection with a Stirling boiler installation. Other features to be noted as conducive to efficiency are: (1) The very large combustion chambers that are provided for retaining the gases at a high temperature to afford sufficient time for combustion before cooling by contact with the boiler tubes. Combustion chambers for bagasse are made larger than for any of the waste fuels previously mentioned. (2) The well-designed baffle walls for mixing the gases

with the oxygen and for retaining a high temperature. (3) The automatic feed hoppers which by means of engine or motor-driven rolls stoke the bagasse at a uniform rate through the furnace roof. From there it falls upon the grate in mounds. There are two

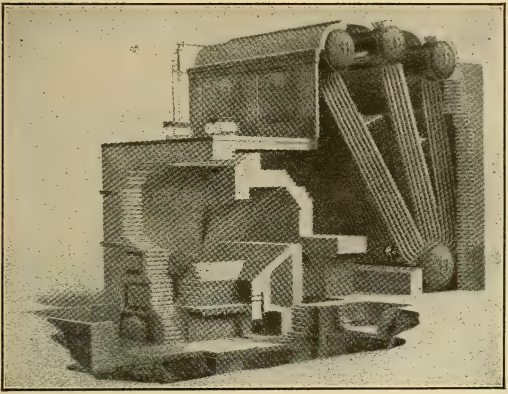


FIG. 44. STIRLING BOILER SET WITH FURNACE FOR BURNING GREEN BAGASSE

of these in the illustration, owing to the large size of the boiler. Ordinarily only one is provided.

Test results show practical agreement with those quoted from Kerr. In neither case, however, was the heat value of the fuel obtained with a bomb calorimeter so that no

accurate deductions can be made as to the absolute efficiency of the furnace and boiler. Again, with this fuel also, the same difficulty in operation arises which has been mentioned in connection with other waste fuels—that is, the trouble which results from having the fuel in the form of cones on the grate, which prevents uniform distribution of the draft through the fuel. Furthermore, the "feast and famine" condition is a very hard one to overcome in the average sugar-mill boiler room. Owing to a little inattention by the firemen the furnaces are allowed to become so full of bagasse that the fires are choked and cooled. This is the feast condition. The famine occurs when the bagasse is not fed fast enough and the grates blow bare in spots around the edges of the heap of fuel and the furnaces are again chilled by the excessive amount of cold air thus admitted.

The principal values given in the test record are as follows:

RECORD OF TEST OF BAGASSE BURNING

Boiler and setting illustrated in Fig. 44

Duration of test, hours	6
Heating surface, square feet	5,750
Grate area, square feet	160
Kind of fuel	Bagasse
Per cent moisture	42.21
Equivalent water actually evaporated from and at 212 degrees, pounds	165,682

Economic Results

Water evaporated per pound of bagasse from and at 212 degrees	2.46
Water evaporated per pound of dry bagasse from and at 212 degrees	4.41
Water evaporated per pound of combustible from and at 212 degrees	4.50

Rate of Combustion

Fuel actually burned per square foot of grate per hour, pounds	70.2
Dry fuel actually burned per square foot of grate per hour, pounds	39.1

Horse Power

Builders' rating in horse power	500
Per cent developed above rating	60

Fig. 45 illustrates a bagasse furnace of the author's that has been designed to reduce these common troubles as far as possible. Two flat fuel beds are provided on the V-shaped grates which eliminate the first difficulty, the feed hoppers being at the sides instead of the centre of the grate. The grate bars are of the type shown in Fig. 46; they produce the opposed blow-pipe effect of draft and flame, and the fuel falls from the side hoppers upon the dead plate for drying and gradually feeds over the grate surfaces as combustion takes place. The "feast" condition is more difficult to obtain with this arrangement as it would be impossible to have

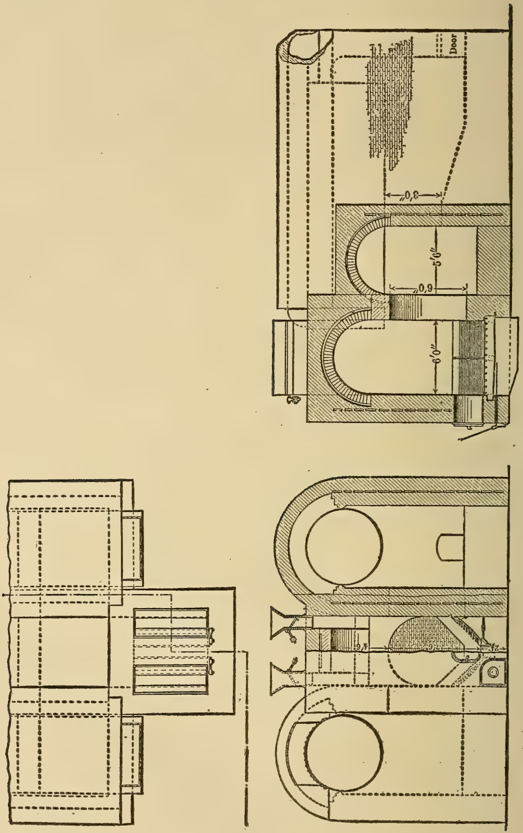


FIG. 45. PLANS AND SECTIONS OF MYERS'S BAGASSE FURNACE

a heap of bagasse rising to the top of the furnace arch, and with the fastest rate of feeding possible there would always be a V-shaped combustion space directly over the bed of fuel. The "famine" condition may occur with any possible design if the fuel supply is reduced below the rate of combustion, and some attention is essential to even fair results.

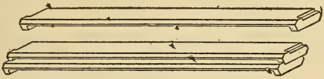


FIG. 46. GRATE BARS FOR MYERS'S BAGASSE FURNACE

Bagasse forms a heavy and troublesome clinker, and a heavy shaking grate is provided at the bottom of the V-grate to reduce the labor and heat losses of hand clinkering. Improvement is much needed in furnace design and furnace control in many bagasse-burning plants. These two factors will often make the difference of running the boiler plant on very little or no purchased fuel, and on the other hand of spending large sums annually for the buying of coal and oil.

CULM

Anthracite culm is, practically speaking, the dust from this coal. The coal from the

mines is screened and graded according to size, each size bearing its own name. The smallest marketable size is No. 3 buckwheat, sometimes called barley. This grade will pass through 3/16-inch and will be retained on 3/32-inch round meshes. Below this there is no grading, and the fine "waste fuel" that falls through the last screen is known as culm. (See Fig. 21.)

This material has constituted until recent years a large waste of fuel. Its very fineness makes it difficult to burn as, in the first place, it opposes the penetration of draft on the grate; and in the second place, under other conditions is caught up by the draft and is blown off up the flue as a dead waste. To emphasize the difficulty surrounding the problem of its efficient combustion, it may be stated that even No. 3 buckwheat demands especially-designed furnaces and scientific treatment in order to obtain economic combustion.

Briquettes

Considerable experimenting has been done in the way of briquetting culm for the purpose of increasing its efficiency. At the present prices of other coals this method is doubtful of success, although a very marked im-

provement in combustion is possible. But the manufacture of briquettes is expensive, requiring special machinery for pressing as well as a binder for maintaining the strength and form of the moulds. The binders that have been tried are flour, water-gas pitch, and residue from heavy oils. The total cost of making briquettes has been from $1.00 to $1.50 per ton, which together with the freight and the cost at the mines makes the delivered price of the fuel prohibitive when used in this way for boiler purposes, at least in most cases.

Some experimenting has also been done in the way of pulverizing anthracite culm to a very fine powder, about 1/125- to 1/200-inch mesh, and then burning it in a brick-lined furnace into which it is blown by a current of air. This has not yet been developed to the stage of a commercial proposition. The method has been successfully used, however, for bituminous coal dust, which ignites more readily owing to its higher content of volatile matter. The fuel burns like a gas in the furnace and has thus been used in connection with cement kilns to considerable extent. A high temperature is obtained and the combustion admits of easy regulation and small labor and is smokeless.

One of the most modern and successful

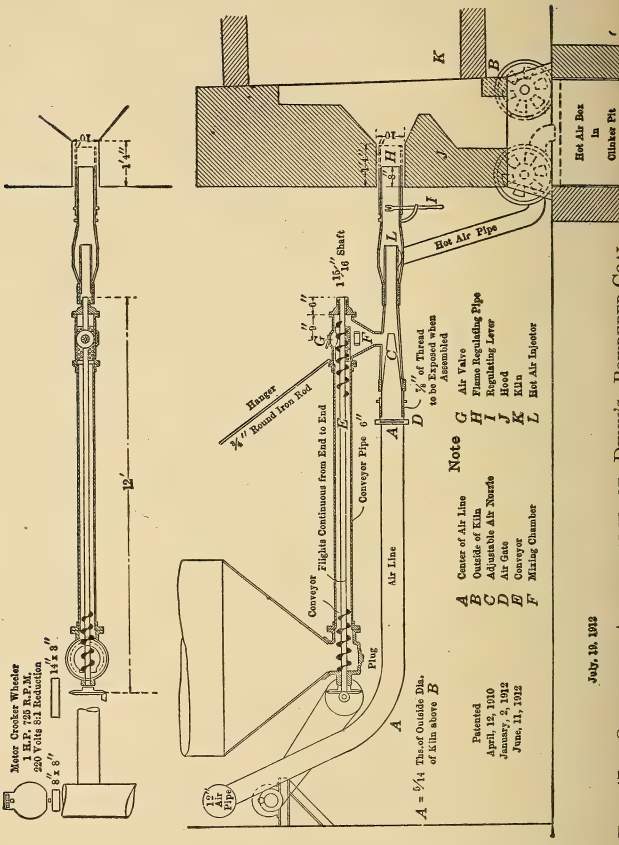


FIG. 47. OPERATING ARRANGEMENT OF DUNN'S POWDERED-COAL BURNER

powdered-coal burners is shown opposite in Fig. 47. This device is the invention of William R. Dunn, and is used most successfully in connection with kilns in the Portland cement industry. It will be noted that the coal in the form of dust is fed by the screw conveyor to a perforated cylinder, from which it is dropped into an annular space surrounding the nozzle "C," which furnishes air at this point under high velocity, thus acting like an injector on the fine coal powder. An auxiliary air pipe is provided which may be supplied with heated air from the clinker pit, means being provided for the accurate regulation and proportioning of air and fuel.

Greater success, and in fact very decided success, has been gained by improvement in design of furnaces for burning culm in its untreated natural condition.

The successful type of furnace is one which combines well-proportioned and controlled mechanical-draft apparatus with properly designed fire-brick arches. The fuel has a low volatile content so that very large combustion chambers are unnecessary, although chambers are sometimes advisable for the purpose of arresting and depositing dust that may be carried over from the fire.

The air is supplied under pressure beneath the grate in a closed ashpit at sufficient in-

tensity to penetrate the bed of culm under all conditions of the fire. The under-grate pressure may be regulated by draft-admission doors to the ashpit. The amount, or volume, of air supplied to the culm is of highest importance and this volumetric regulation is effected principally at the damper in the uptake of the boiler. This damper is frequently controlled by an automatic regulator which partly closes and opens as the steam pressure in the boiler rises and lowers. But the damper must first be set by hand to give the correct throttling effect between the determined extremes of operation of the boiler. Furthermore, care must be taken to increase the rate of firing as the damper opens wider, and to reduce the feeding of fuel as the draft decreases. If these precautions are not followed, the damper regulator may operate to reduce the air flow for a heavy fire and to increase the air for a light fire, thus destroying instead of improving the economy. In other words, the usual form of automatic damper-regulator works as a function of the varying steam pressure, which latter depends *not alone* on the rate of combustion but as well upon the rate of steam flow from the boiler. This is a point rarely mentioned, if recognized, by promoters of this class of damper-regulators.

Control of the under-grate draft is frequently effected by means of the same class of damper regulator as above referred to. In this case, however, a balanced valve is inserted in the steam line to the fan engine, or in the supply pipe of a steam blast which is often used to create air supply under pressure.

This balanced valve is connected to the damper regulator and by this means the supply of steam to the fan or blower is regulated in accordance with variations of the steam pressure at the boiler. The same precautions are necessary with this contrivance as have been indicated in connection with the opening and closing of the damper when the control of the draft is effected in that way. That is to say, when the draft increases, an increase in the rate of firing must also be made, and vice versa, for the purpose of maintaining the air and fuel supply in proper relation to each other.

A well constructed set of dumping grates for disposing of the clinker, and also the admission of steam under the grates for softening this clinker, are valuable adjuncts to a culm-burning furnace. A careful study of conditions is essential to determine the economic ratio of grate surface to the heating surface of the boiler.

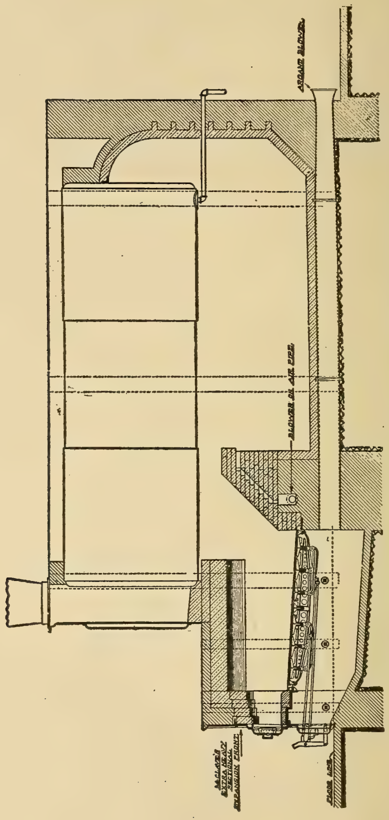


FIG. 48. McCLAVE CULM-BURNING FURNACE

The notable advance in the utilization of anthracite culm on a large scale is indicated by the plant of the Harwood Electric Company, Harwood, Pa., where twenty-two 950-horse-power boilers are installed to operate on this fuel which costs them about five cents per ton, delivered. By burning the culm directly at the mines, the cost of freight is eliminated and cheap electric current is generated for high-tension distribution through the surrounding territory.

Another similar plant is planned for Mauch Chunk, Pa. The entire supply of culm is to be contracted for at the rate of a few cents per ton. The ash and clinker are to be sold back to the mines for filling-in purposes, which still further reduces the net cost of fuel. The culm is to be burned in special furnaces designed along the lines described, and briquetting will not be necessary.

Thus a valuable by-product which was formerly considered a mere waste and great detriment and was piled up into the enormous culm heaps familiar to the anthracite region, has at last by virtue of scientific furnace design come into its own to fill an important place in the great economy of power generation.

Fig. 48 illustrates a culm-burning furnace and boiler setting, using McClave-Brooks

dumping grates and fire-brick arches; the latter for the double purpose of maintaining ignition temperature over the fire and for checking the flight of dust particles that may be caught up by the draft.

The following data in Test B are taken from a complete test on a McClave equipment, while Test A shows the comparatively poor results on a furnace without an arch over the fire.

Location of Plant, Reading, Pa. *Kind of Boiler, H.T.* *Fuel, River Coal Refuse*	*Test A Regular Setting*	*Test B Projected Furnace McClave Design*
Ratio grate to heating surface	1 to 50	1 to 40
Water-heating surface, sq. ft.	1,949	1,949
Per cent moisture in coal...	9.06	10.63
Dry fuel burned per sq. ft. grate surface per hour....	16	15
Horse power developed.....	158.7	202.76
Equivalent evaporation from and at 212 degrees per lb. coal as fired............	7.96	8.6
Equivalent evaporation from and at 212 degrees per lb. dry coal..................	8.75	9.633
Percentage of ash in dry coal	......	......
Percentage of volatile in dry coal....................	......	......
Calorific value of the dry coal per lb., B.t.u........	12,724	12,178
Efficiency of boiler, including furnace (based on dry coal), per cent..........	66.4	76.39

The culm used in these tests makes a striking example of the reclaiming of waste fuel, for the material used is known as "river coal refuse." This consists of "tailings" which compose the waste from the screening and washing, and which are carried into the river. These tailings are made up chiefly of culm under 3/32-inch mesh with some small proportion of coarser particles. This coal dust settles where it may on the river bottom and is dredged up for use under boilers equipped with special furnaces for its combustion, a true example of reclamation.

An inspection of Fig. 48 will show the special method of preheating air over the furnace arch and of introducing this air under pressure in such a manner as to cause a forced intermingling or mixing action in the furnace with the fuel gases. Another heated-air admission to still further increase this effect and to cause a whirring or baffling action is shown at the bridge wall of the furnace. The blast is caused by steam blowers which also reduce and soften the formation of clinker. Dumping grates of special construction facilitate the cleaning of fires.

A Parson system culm furnace is shown in Fig. 49. This also is of the Dutch-oven type and the draft is furnished by steam-blast apparatus. An automatic draft regula-

tor of the general type previously referred to is shown as a part of this equipment, and a secondary air supply is provided over the

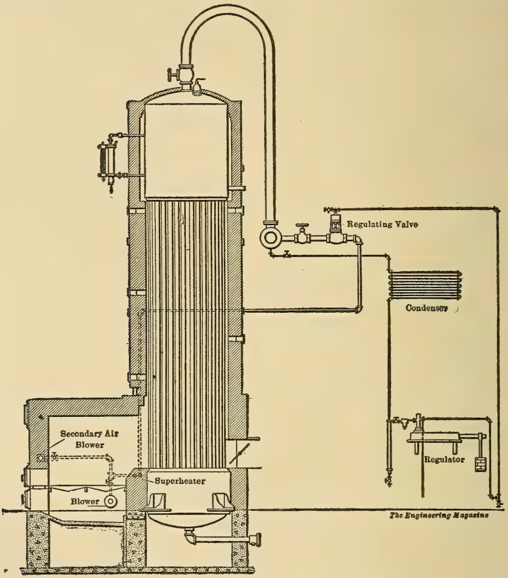


FIG. 49. CULM-BURNING FURNACE ON THE PARSON SYSTEM

fire. The Parson Co. kindly furnished the following test made under *forced conditions,* which illustrates the combustion of a fresh-mined culm consisting of 13.55 per cent rice, 44.8 per cent barley and 41.65 per cent

"dirt," the last constituent indicating, no doubt, fine dust.

Test on Parson System

Fuel used	Fresh-mined culm
Ratio of grate surface to heating surface	1 to 37.3
Water-heating surface, sq. ft	3,030
Per cent moisture in coal	3
Dry coal burned per sq. ft. grate surface per hour	22.26
Horse power developed	447.3
Equivalent evaporation from and at 212 degrees per pound of coal as fired	8.35
Equivalent evaporation from and at 212 degrees per pound of dry coal	8.61

Coke Braize

When gas is made by destructive distillation of soft coal, which is the older method employed for city gas production, there is left as one of the important by-products a finely divided coke. The volatile gases have been driven off, so that the coke that remains is principally carbon which is naturally high in ash. The particles are about the size of No. 2 or No. 3 buckwheat coal, and it can be successfully used as a boiler fuel with proper furnace equipment. It is capable of competing with the fine sizes of low-priced anthracite coal.

Coke braize is best burned with an undergrate draft pressure of considerable intensity. A draft pressure of 1-inch water gauge was used in a test made by the author. This draft was supplied by a steel-plate fan delivering into the closed ashpits of two water-tube boilers. Some of the results and data are as follows:

TEST OF BURNING COKE BRAIZE

Cost of coke braize per long ton delivered....	\$2.30
Percentage of ash........................	21
Evaporation per lb. of coke braize from and at 212 degrees, lb........................	6.25
Evaporation per lb. of combustible from and at 212 degrees, lb......................	7.53
Cost to evaporate 1,000 pounds of water into steam from and at 212 degrees............	\$0.164
CO_2—average in flue gases, per cent........	10.25

The principal characteristics developed in burning this fuel are:

1. A very hot fire with long flame.
2. Formation of a large amount of very hard and troublesome clinker.

It was found by experiment that by making the coke braize very wet before firing results were much improved. The moisture softened and reduced the clinker materially. Better results could undoubtedly have been obtained with further practice.

City Refuse

City refuse belongs to the class of fuels which, by application of the scientific principles of combustion, have been promoted from the state of semi-combustible into that of auto-combustible fuel.

While this heterogeneous material can be disposed of by dumping at sea or on waste land—sometimes in an inexpensive manner—the consensus of opinion is strongly in favor of its disposition by combustion. The strength of this opinion is largely due to comparatively recent improvements made in the design and efficiency of destructors or refuse burners. Furthermore, when combustion is complete, burning is by far the most sanitary method of disposing of this and other offensive materials.

It is a notable achievement of engineering study and skill that has made possible not only the smokeless and complete combustion of city refuse, but also the production of useful heat and power from this former waste.

Some of the collected matter, principally the house garbage, contains so much moisture in comparison with its low heating value that to burn it alone would be impossible. But by mixing this with all the other matter,

the calorific value of the mass is made sufficiently high not only to evaporate its contained moisture but to produce effective evaporation in a boiler as well. Even when this is done, however, it is necessary to depend for efficiency upon scientific design and careful operation of the furnace.

To gain an idea of the composition of city refuse, a quotation is herewith given from the acceptance tests made on a Heenan-Froude destructor in operation on Staten Island, N. Y., at West Brighton. This is an English design of burner and the tests were conducted for the city by their engineer, Mr. Fetherston, to whom the author owes his acknowledgments.

Test of Garbage Burning, Heenan-Froude Destructor

	Test No. 1[1]	*Test No. 2*[2]
Garbage[3]	46.6	11.8
Fine ash	21.7	
Coal and cinders	7.7	79.5
Clinker	0.6	
Glass and metals	8.5	3.4
Rubbish	14.9	5.3

[1] Sept. Mixture. Prepared artificially.
[2] Feb. Mixture. Prepared artificially.
[3] Garbage consists of animal and vegetable matter.

	Test No. 1	*Test No. 2*
Date, 1908	May 6	May 13
Duration, hours	8	8
Material (see above notes)	Sept. Mix.	Feb. Mix.
Evaporation per lb. of refuse burned, gross actual lb.	1.17	1.10
Net useful steam for power purposes from and at 212 degrees, lb	1.31	1.24
CO_2 average per cent	12.2	12.5
Temperature of chimney gases	393	364

The capacity of this burner is 60 tons per day and is the first of this design installed in

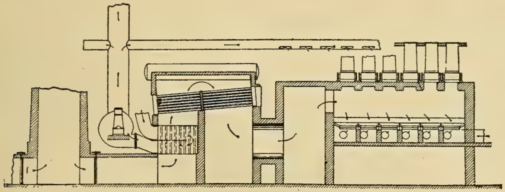


FIG. 50. GENERAL DESIGN OF THE HEENAN-FROUDE REFUSE DESTRUCTOR

the United States. A cut showing the general design is given in Fig. 50. Fig. 52 (page 461) shows an exterior view of the West Brighton Plant.

Some of the features of especial note are

the very large combustion chambers, the preheated air supply delivered under pressure below the grates, and the division of the space below the grate so that the draft can be shut off for any one section while clinkering.

An additional feature has been successfully attained by passing the air supply for the furnace through the white hot clinker, which performs the double function of cooling the clinker for handling and of reclaiming much of its heat for the improvement of the furnace efficiency. Mr. Fetherston has done valuable work in this connection. The exhaustive tests made by him for the City are recorded in Vol. LX of the American Society of Mechanical Engineers.

Colonel Wm. F. Morse has made valuable contribution to this field of municipal waste disposal through his deep study and important book on the "Collection and Disposal of Municipal Waste," through his own design of destructors, and through his experience in the handling of such problems.

Colonel Morse acts also in consulting capacity in connection with the construction in this country of the Sterling destructor. This is also an English furnace and it is largely used in European and other countries. Fig. 51 gives an idea of the design of this burner.

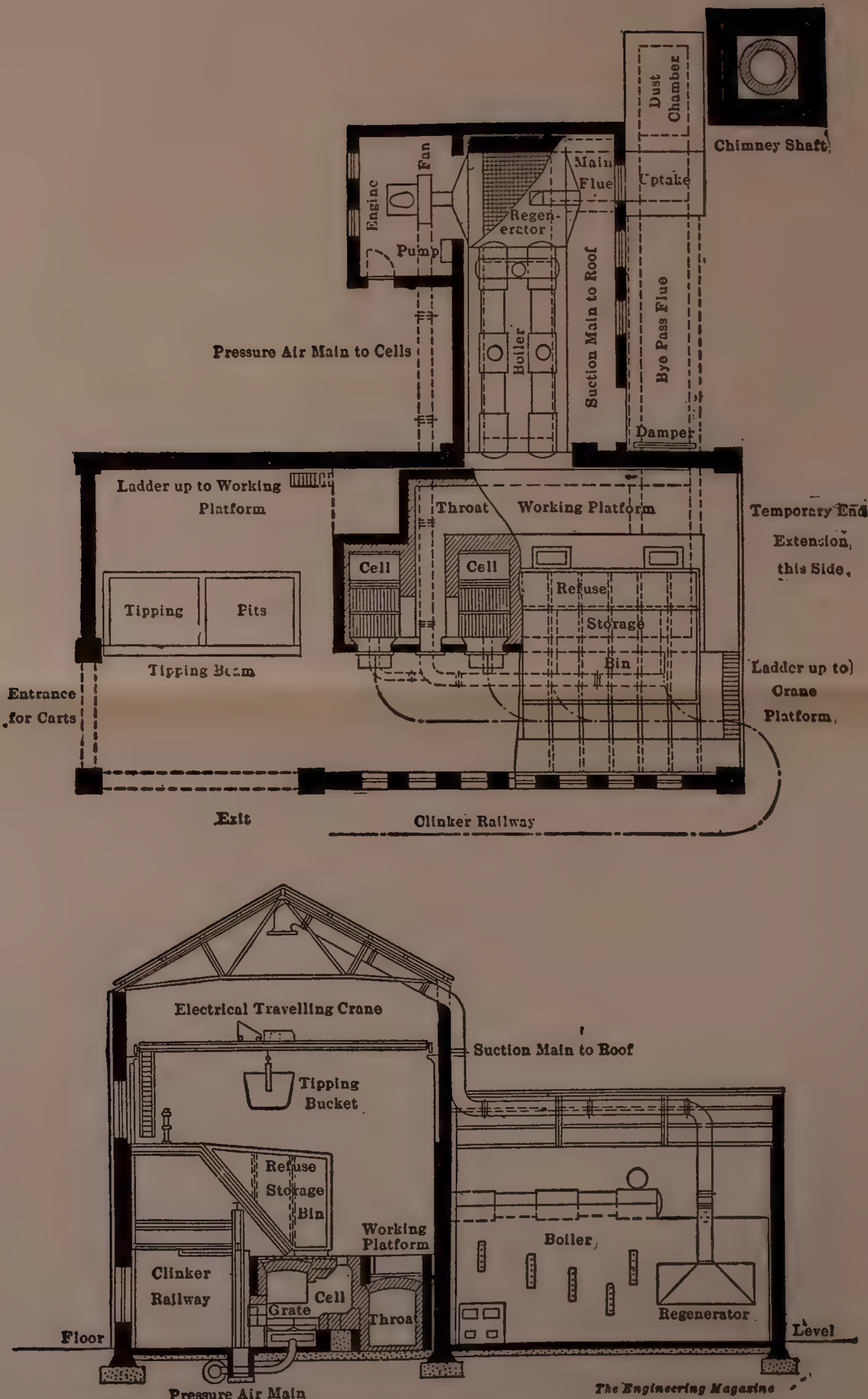


FIG. 51. PLAN AND ELEVATION OF THE STERLING REFUSE DESTRUCTOR

One feature of especial importance for this fuel, used by various destructor companies, is shown clearly on this cut, i. e., the drying hearths or dead plates. In charging, the wet fuel is placed upon these where it is made

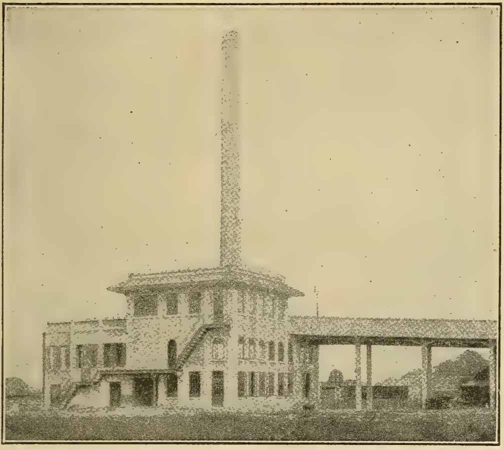


FIG. 52. THE HEENAN-FROUDE REFUSE DESTRUCTOR AT WEST NEW BRIGHTON

sufficiently dry for burning before being raked over onto the real grate surface.

Evaporative results vary with the composition of the refuse, but 1¼ pounds of steam from and at 212 degrees per pound of refuse appears to be a usual figure for guarantees

when "average" refuse is supplied. Considerably higher results have been obtained in the official tests recorded. An English Sterling plant officially reports, as an average for one year, an evaporation per pound of refuse destroyed of 1.4 pounds of water into steam.

The clinker that results from the burning of city refuse is crushed, combined with Portland cement, and formed into concrete blocks useful for building purposes. These are sold and thus reduce the cost or increase the profit of operation of the plant.

In further relation to composition of city refuse, Mr. George Watson in a paper before the Institution of Mechanical Engineers states that "it is a fairly safe generalization to say that in England it consists of one-third by weight of water, one-third combustible matter, and one-third incombustible." This last is the portion withdrawn in the form of clinker to be crushed and formed into blocks for building or other purposes.

In the early days of refuse destructors they were operated entirely on natural chimney draft. A partial vacuum was thus always present in the furnace. Since the clinkering and charging operations required that the furnace doors should remain open a considerable time, very great volumes of cold

air would enter the furnace by virtue of the higher pressure of the atmosphere outside.

To Mr. William Horsfall is credited the correction of this cause of inefficiency by the introduction of forced draft under the grates. Thus by proper regulation the pressure in the furnace becomes equal to the atmospheric pressure outside, so that the detrimental inrush of cold air while the doors are open is prevented, and the efficiency of combustion is largely increased.

Another design of refuse destructor which was recently invented in this country by Dr. J. B. Harris is shown in Fig. 53. Provision is made for drying out the wet bulky material on water grates above the fire, these grates being formed by the upper tubes of a boiler constructed within the furnace. It is claimed that carcasses of dead animals can thus be effectively incinerated.

Certain essential features are common to the most successful designs of refuse destructors. They may be summarized as follows:

1. Very large combustion spaces or chambers are provided for checking the velocity of the gases and affording time for their diffusion with the air for combustion.

2. The fuel is surrounded by hot firebrick surfaces, the walls of which are capable

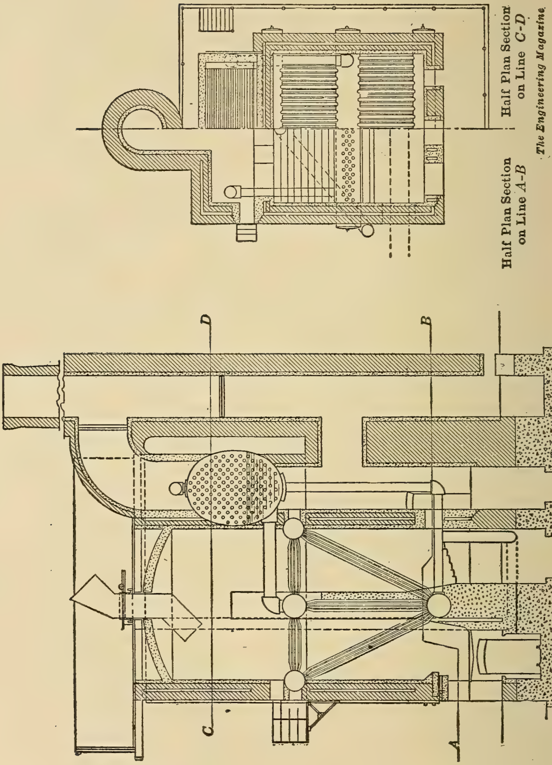


FIG. 53. REFUSE DESTRUCTOR INVENTED BY J. B. HARRIS

of retaining sufficient heat to gasify and cause the ignition of the fresh charge of fuel.

3. The fresh charges are small in comparison with the mass of burning refuse, in order to prevent the serious lowering of the furnace temperature. Necessarily, therefore, the charging or firing is done at short intervals.

4. An under-grate preheated forced draft of high intensity is employed to obtain (a) balanced furnace conditions, (b) an effective penetration of the mass of fuel, and (c) controlled combustion.

5. Drying hearths or dead plates or other means are provided for evaporating the moisture from the refuse before it reaches the grate surface.

6. A very important feature in operation is the mixing together of all the different classes of refuse to be destroyed. This produces a constant supply of fuel of the average calorific power for the furnace. Otherwise it would be easy to "kill" the fire by suddenly introducing a quantity of very wet or very bad refuse.

7. Between one and two pounds of steam are generated for each pound of refuse destroyed, depending upon the composition of this refuse and upon the efficiency of design and operation of the destructor.

8. It has been found best from the standpoint of efficiency as a general rule not to sort and pick the refuse but to let all go to the destructor. The picking of refuse is unsanitary for the workers and dangerous, owing to possibility of contagious disease.

There are still other substances which could be treated as waste fuels, but their discussion would, in the mind of the reader, merely emphasize the variety of methods used to apply the same principles of combustion.

In summing up this broad subject of waste fuels, the author can only refer once more to the three simple requirements of combustion which apply to all fuels.

1. High temperature.
2. Correct air supply.
3. Complete mixture of this air with the fuel gases.

These combustion requirements are common to all fuels—rich or poor in heat values, good or bad in physical or chemical composition. The solution of any specific problem, the burning of any specific fuel, depends upon the scientific application of these common requirements to the particular material in question. This belongs to the field of scientific design and operation of furnaces. That much advance has been accomplished in

recent years in this field is demonstrated by the production of useful power-generating combustion with matter that had previously been considered either worthless as fuel or as belonging to the semi-combustible class which required the burning of expensive fuel for its ignition.

It must be remembered, however, that such fuels intrinsically contain more heat units than are actually required for the maintenance of their gasification and ignition. If they did not, to burn them would be as impossible as to make a stream run up hill. If they do contain a sufficient excess of heat over and above the gasification and ignition requirements, then the efficiency of their combustion depends entirely upon the design and operation of the furnace.

Many quotations are made in text and reference books as to the heat value of moist and fibrous fuels which are entirely misleading if not useless. This is owing, in the first place, to the variety of ways in which heat values may be reported and the absence in such quotations of definite specifications as to which of four different methods is used. Further—and equally serious—error may arise from dependence for calorific powers on formulas like Dulong's, which assume that all of the oxygen in a fuel is combined

before combustion with the hydrogen in that fuel. The error may be slight in fuels containing but little oxygen, but increases to serious extent with woody and many other waste fuels high in oxygen. Calculations containing this common error are unsafe and useless, both for comparison of combustion performances and for predicting results in any problem dealing with these fuels.

As a final statement concerning the successful treatment of waste fuels, perhaps the most specific and at the same time the most general conclusion would be the following:

As physical human beings are governed by the laws of hygiene common to all, so all waste fuels must be treated under the nature-determined laws of combustion common to fuels of every description; and likewise, there being no one medicine or treatment that will give perfect health to all persons, each individual requiring specific treatment according to his special condition, so also each particular waste fuel constitutes a specific problem of its own and requires special study and treatment according to its own peculiar chemical and physical characteristics.

Chapter XVII

BOILER-FEED PUMPS; STEAM CONSUMPTION TESTS

FOR supplying water to the boilers the direct-acting type of steam pump is almost universally employed.

In the design of an efficient power plant the amount of exhaust steam to be produced must be estimated, and yet there are but few data in the form of actual tests on feed pumps available for such use. The most convenient form of expressing the steam consumption of a feed pump is in a percentage of the total steam developed by the boilers which are served by the pump in question, and I have made careful tests to determine this value under working factory conditions in two places. These tests are given below.

FEED-PUMP STEAM CONSUMPTION

Test No. 1, June 3, 1913

Make of pump	Snow
Type of pump	Duplex

Size of pump	$6 \times 4 \times 6$
Size of feed line	2 inch
Duration of test	1½ hrs.
Exhaust from pump condensed in tank and condensation measured.	
Steam pressure in boiler	115 lb.
Number of boilers served	2
Temperature of feed water (approximate)	185 degrees
Water fed by pump to boiler (total)	8,330 lb.
Steam condensed from pump exhaust, total	552 lb.
Per cent of boiler output used by feed pump, 552 ÷ 8,330	6.63
Boiler horse power developed at 30 lb. steam per horse power per hour	185
Boiler horse power used by pump @ 30 lb. steam per horse power per hour	12.25

Conditions

This was an old pump, its exact age not being known. When the valve on the discharge line was closed and full steam pressure admitted to the steam cylinders the pump showed a very small amount of slip in the water end and a comparatively small amount of leakage past the steam valves and the steam piston, all of which indicate that the pump was in average working condition when tested. Consequently this test may be taken as a fair indication of the amount of steam consumed by a duplex feed pump when handling two boilers under conditions of operation corresponding to those existing in this plant.

It should be further noted that the pump was being regulated by a feed-water regulator and not by hand. The regulator when off did not quite stop the action of the pump, but allowed it to run very slowly so that it would scarcely discharge any water to the boilers during the off-time of the regulator. The pump would run about 1 minute or 1½ minutes at full speed and then the regulator would be off for about two to three minutes.

Another condition of operation which should be noted is that the pump was taking cold water from measuring barrels having close connection to pump and forcing this water through a short length of pipe and thence through a Reilly multicoil heater and from there to the boilers with an average number of turns in the feed piping which was 2-inch.

Method of Testing

During the test all water fed to the boilers was measured in a feed-water weigher tested to within ⅓ of one per cent accuracy. The exhaust pipe of the feed pump was turned directly below the surface of cold water in a calibrated tank and so condensed. The weight of condensation was calculated from the increase in volume of the water, with

proper consideration of volume change due to the increased temperature at the end of the test.

FEED-PUMP STEAM CONSUMPTION

Test No. 2—July 10, 1913

Make of pump	......
Type of pump	Duplex hand reg.
Size of pump	4½×3×5
Size of feed line	......
Duration of test	2 hr. 37 min.
Steam pressure in boiler	50 lb.
Number of boilers served	2
Temperature of feed water	68 degrees
Water fed by pump to boiler	11,700 lb.
Water fed by pump to boiler per hour	4,475 lb.
Steam condensed from pump exhaust, total	684 lb.
Per cent of boiler output used by feed pump	5.84
Boiler horse power developed at 30 lb. steam per horse power per hour	149
Boiler horse power used by pump @ 30 lb. steam per horse power per hour	8.7

Conditions

This pump was an old second-hand one. It took water at 68 degrees and discharged it through a Berryman closed feed-water heater and about 50 feet of pipe into two horizontal tubular boilers. The general conditions of piping, number of turns, and general oper-

ation might be called average for small factory power plants. The pump was regulated by hand, and drew its supply from a sump tank or barrel, the water level in which was about 18 inches above the level of the pump suction.

Method of Testing

The method of testing was precisely the same as that employed in test No. 1. (See pages 469, 470.)

The steam consumption of a feed pump will depend upon the mechanical and thermal efficiency of the pump itself under the imposed conditions of steam pressure, back pressure, and load, and upon the total head under which it works. These conditions will vary considerably, but a direct-acting steam pump is at best the most inefficient type of steam motor, and should be used only where steam economy is of secondary importance or where the exhaust steam can be thoroughly utilized, as for instance in a feed-water heater or in the steam coils of a heating system.

In any event the foregoing tests have their peculiar value as being based on no theoretical considerations whatever, but upon actual performance under the stated conditions; and these results will serve well as a basis of

computation for the amount of exhaust to be cared for in the layout of many small-sized factory plants. In such it may be safely figured that from 5 to 7 per cent of the total boiler steam will be required to operate the feed pump when the latter is of the direct-acting duplex type. The single-cylinder steam pump will take a little less steam than the duplex, owing to slightly less cylinder condensation, a less percentage of clearance, and less friction of pistons, rods and valves. The single pump on the other hand in usual designs is not so nearly positive in the actuation of its steam valves.

The following calculation of efficiency will prove interesting as applied to the pump of test No. 1, this one being selected as being representative of a better average condition as to steam pressure on the boilers, i. e., 115 pounds per square inch.

Assuming the total head against which the pump was working to have been 120 pounds (allowing for lift to boilers and friction of piping, elbows and valves), the theoretical horse power of pumping would be

$$H.P. = \frac{W.H.}{33{,}000},$$

in which

W = Weight of water per minute = 92.55
H = Height of lift = 120×2.3094 = 277.1 ft. hd.

$$\text{Theoretical horse power} = \frac{92.55 \times 277.1}{33,000} = 0.777$$

horse power

That is, the actual work of moving the water at the above rate without loss was 0.777 horse power. From the pump test 552 ÷ 1.5 = 368 pounds of steam per hour were actually required to do this work. Hence the steam consumed per utilized horse-power hour was 368 ÷ 0.777 = 473 pounds.

Now if the mechanical efficiency of the pump is 50 per cent (see Kent) the indicated horse power of the pump was 0.777 ÷ 0.50 = 1.554 indicated horse power and the steam consumption of the pump per indicated horse power per hour was 237 pounds.

A pound of steam at the boiler pressure contained 1,191 B.t.u. above 32 degrees F. and 237 pounds would contain 282,267 B.t.u. The heat equivalent of a mechanical horse-power hour is 2,545 B.t.u. Therefore the thermal efficiency of the pump referred to indicated horse power would be 2,545 ÷ 282,267 = 0.9 per cent.

The thermal efficiency referred to the energy actually utilized in moving the water would be one-half of this figure, i. e., 0.45 per cent. The efficiency of an engine using 30 pounds of steam at this pressure referred to

actual or brake horse power would be 2,545 $\div (30 \times 1{,}191) = 7.12$ per cent.

Therefore we may say that the pump operated at an efficiency of 0.1263 or about ⅛ of the above engine efficiency when referred to indicated horse power, and at about 1/16 of this engine efficiency when referred to utilized energy.

The following test on the steam consumption of a duplex feed pump made by Mr. S. Milton Clark, M. E., is interesting as additional data checking closely the results obtained by the author on the two other tests quoted in this chapter. The conditions of this test were the same as those prevailing in the other tests on feed pumps quoted herewith. That is to say, the water fed to the boiler was obtained by actual weights and the exhaust from the pump was condensed and weighed.

FEED-PUMP STEAM CONSUMPTION

August 30, 1912

Make of pump	Fairbanks-Morse
Type of pump	Duplex
Size of pump	$6 \times 4 \times 6$
Size of feed line	......
Duration of test	7 hours
Steam pressure in boiler	89.06 lb.
Number of boilers served	1

Temperature of feed water	185.2 degrees F.
Water fed by pump to boiler	19,520 lb.
Steam condensed from pump exhaust, total	1,145 lb.
Per cent of boiler output used by feed pump	5.87
Boiler horse power developed at 30 lb. steam per horse power per hour	93
Boiler horse power used by pump @ 30 lb. steam per horse power per hour	5.45

Chapter XVIII

MODERN TYPES OF PRIME MOVERS

THE old-fashioned portable slide-valve engine mounted upon its own boiler and used largely for driving small temporary saw-mills and for similar light service consumed in the neighborhood of 8 to 10 pounds of coal or its equivalent per brake horse-power hour. Fig. 54 illustrates this familiar device.

Modern designs of engines mounted upon their boilers in Germany are recorded to have produced a brake horse-power hour for less than one pound of coal. Messrs. R. Wolf of Magdeburg, Germany, may be said to be the original developers of this type of unit, known on the continent as the "locomobile." Other European makers are now supplying the increasing market for these remarkable machines. It is stated that in Germany one firm alone has built nearly 1,000,000 horse power of these units.

The Locomobile

The locomobile consists essentially of a very efficient type of condensing or noncondensing engine mounted upon an internally

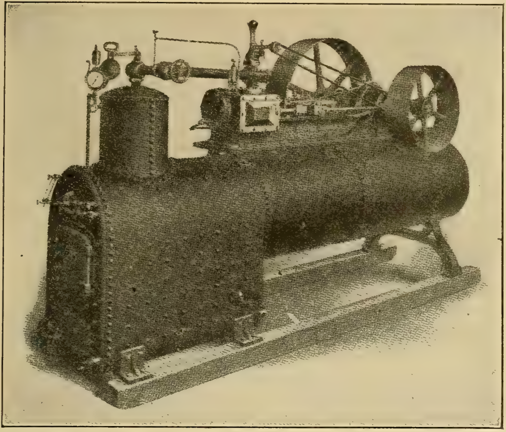


Fig. 54. Regular Type of Portable Engine

fired high-pressure boiler generating highly superheated steam. The compound-engine cylinders are jacketed in the flow of the hot flue gases on their way to the stack, to reduce cylinder condensation and radiation. The exhaust steam from the high-pressure cylinder passes through a special superheater

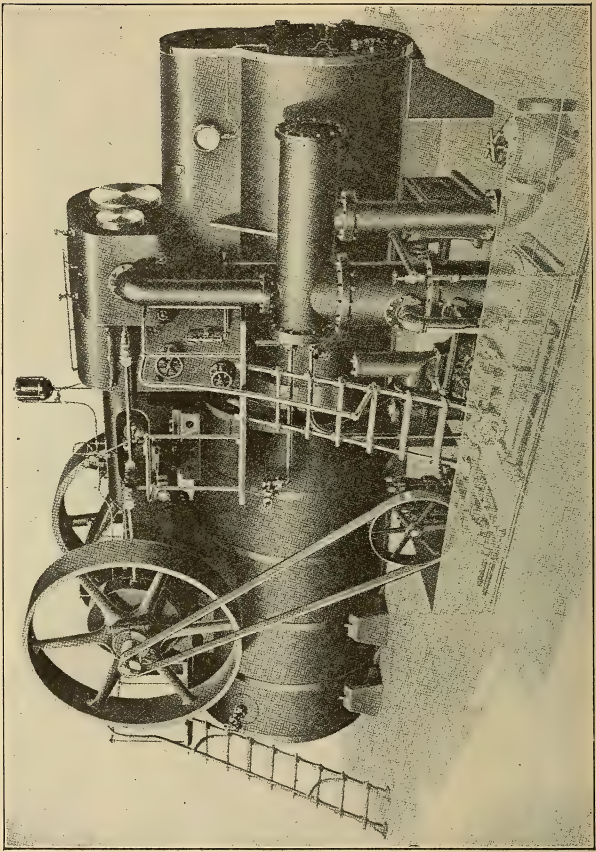


Fig. 55. The Buckeyemobile

before returning to the low-pressure cylinder. A closed feed-water heater returns some of the heat of the exhaust steam to the boiler.

A locomobile type of self-contained unit is now manufactured in the United States and is known as the "Buckeyemobile." It is made in sizes from 75 horse power to 600 horse power. Fig. 55 shows this machine in the tandem compound condensing design. Fig. 56 shows the internal parts and gives a clear idea of their operation.

As regards low fuel consumption, the locomobile type of steam plant competes on an equal basis with producer-gas plants and has in its favor the added advantage of the greater reliability of steam-driven as compared to gas-driven engines.

It is further of interest to note that the high over-all efficiency of the locomobile type of plant is obtainable *in spite of an ordinary, rather than with the aid of a high, boiler and furnace efficiency.* If we could apply the Bone system of surface combustion (with oil or gas) to the locomobile, its over-all efficiency[1] would be increased by more than 18

[1] Efficiency of locomobile boiler taken at 70 per cent which would represent favorable conditions of coal and firing. Efficiency of Bone boiler taken at 83 per cent—see chapter on Surface Combustion. Then gain of over-all efficiency would be $\frac{83 - 70}{70} = 18.6$ per cent, and the saving of fuel for the same output would be $\frac{83 - 70}{83} = 15.7$ per cent.

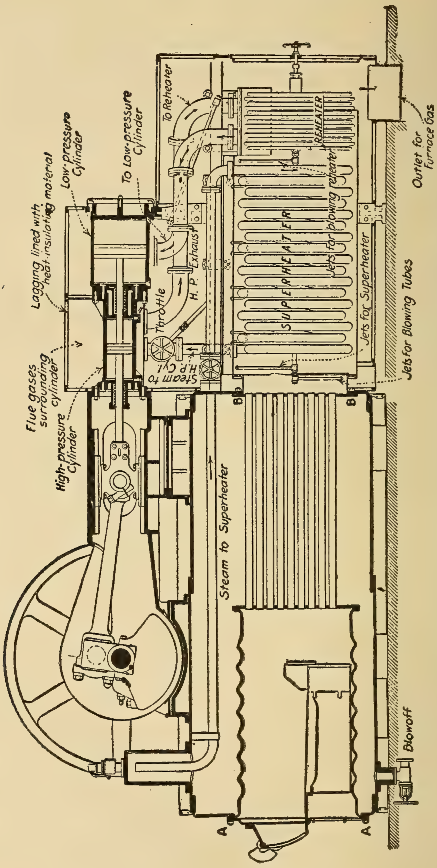


FIG. 56. LONGITUDINAL SECTION OF THE BUCKEYEMOBILE

per cent and for the same output a saving of fuel (heat units) amounting to nearly 16 per cent would result. Moreover, the capacity of the locomobile would be multiplied by about *five,* so that it would be practicable to increase the size of the prime mover to a point where a steam turbine would give economy equal to that of a reciprocating engine. It is altogether likely that development along these lines will take place with a view to use in localities where oil or gas is available. The further extension of this idea for *coal* burning depends upon the questions outlined in the chapter on Surface Combustion.

These considerations are purely prophetic in character, but we need not anticipate future improvements in order to realize that the small 400 horse-power to 600 horse-power locomobile units as now constructed in this country equal the performance in low fuel consumption of the most refined and largest central steam-power stations ever designed. In fact, these small units are able to show coal economies which exceed the results regularly obtained in highly refined central stations of 50 to 200 times their capacity.

Thus a small manufacturer may generate his own current at the same fuel expense per kilowatt hour as it costs the public utility corporation. This is omitting any consider-

ation of the use of exhaust steam when condensing water is available.

The following test on a 150 horse-power Buckeyemobile was made by Professor Thos. G. Estep of the mechanical-engineering department of Carnegie Institute. In the examination of this test it should be remembered that it was on one of the smaller size machines and that still better economy would be expected on a larger unit.

OBSERVED AND CALCULATED RESULTS OF TEST

General

Total heating surface of boiler, sq. ft....	350.0
Heating surface in service during test, sq. ft....	250.0
Grate surface, 2 ft. 6 in. by 3 ft. 0 in., or sq. ft....	7.5
Ratio of grate surface to heating surface.	1 to 33.3
Superheating surface, sq. ft....	257.0
Reheating surface, sq. ft....	96.0
Size of engine....	8½ in. by 17 in. by 18 in.
Rated horse power of engine....	150.0
Duration of trial, hours....	8.0
Kind of fuel.....Low-grade Pocahontas run-of-mine	
Kind of draft....	induced

Average Pressures

Steam pressure in boiler lb. per sq. in. gage....	220.0
Barometer, inches of mercury....	28.85
Absolute steam pressure, lb. per sq. in...	234.1
Receiver pressure, lb. per sq. in. gage....	9.0
Absolute receiver pressure, lb. per sq. in..	23.1

Vacuum referred to 30 in. barometer, inches of mercury	25.0
Absolute condenser pressure, inches of mercury	5.0
Absolute condenser pressure, lb. per sq. in.	2.45
Draft in furnace, inches of water	0.10
Draft in breeching, inches of water	0.31

Average Temperatures, in Degrees Fahrenheit

Temperature of steam at throttle	595.0
Saturation temperature at throttle	395.2
Superheat at throttle	199.8
Temperature high-pressure exhaust	271.0
Saturation temperature of high-pressure exhaust	235.7
Superheat in high-pressure exhaust	35.2
Temperature of steam leaving reheater	367.0
Superheat to low-pressure cylinder	131.3
Temperature of low-pressure exhaust	132.0
Saturation temperature of low-pressure exhaust	132.0
Superheat in low-pressure exhaust	none
Temperature of feed water entering heater	108.0
Temperature of feed water leaving heater	126.0
Temperature of gases leaving boiler	736.0
Temperature of gases in breeching	419.0
Temperature of injection water to condenser	96.0
Temperature of overflow from condenser	114.8
Temperature of fire room, approximately	80.0

Flue-Gas Analysis, in Percentage by Volume

Carbon dioxide	11.3
Carbon monoxide	0.24
Oxygen	5.6
Nitrogen	82.86
	100.00

Miscellaneous Averages

Speed of engine, revolutions per minute	225.0
Indicated horse power in high-pressure cylinder	74.1
Indicated horse power in low-pressure cylinder	55.6
Total combined indicated horse power	129.7
Brake horse power	125.5
Mechanical efficiency of engine, per cent.	96.8

Proximate Coal Analysis

Moisture, per cent	2.21
Volatile matter, per cent	15.55
Fixed carbon per cent	69.89
Ash, per cent	12.35
Sulphur, per cent	0.48

Ultimate Analysis

Carbon, per cent	78.06
Hydrogen, per cent	3.88
Nitrogen, per cent	0.94
Oxygen, per cent	3.98
Ash, per cent	12.64
Sulphur, per cent	0.50
B.t.u. per pound of dry coal	13,541

Total Quantities

Total water fed to boiler, lb	11,430
Moisture in steam assumed, per cent	1.5
Quality of steam, per cent	98.5
Total water actually evaporated by boiler, lb	11,258.5
Factor of evaporation, boiler alone	1.1403
Factor of evaporation, boiler and superheater	1.3015
Equivalent evaporation from and at 212 degrees F., by boiler alone, lb	12,838
Equivalent evaporation from and at 212 degrees F., by boiler and superheater, lb	14,653
Total coal as fired, lb	1,397
Moisture in coal, per cent	2.21

Total dry coal used, lb	1,366.1
Ash from test, lb	87.0
Ash from test, per cent	6.37
Ash by analysis, per cent	12.35
Total combustible (based on ash from analysis) lb	1,197.4

Hourly Quantities and Rates

Water fed to boiler per hour, lb	1,428.8
Water actually evaporated by boiler per hour, lb	1,407.3
Equivalent evaporation from and at 212 degrees F. per hour by boiler alone, lb.	1,604.8
Equivalent evaporation from and at 212 degrees F. per hour, by boiler and superheater, lb	1,831.6
Boiler horse power developed by boiler alone	46.6
Boiler horse power developed by boiler and superheater	53.2
Coal as fired per hour, lb	174.6
Dry coal per hour, lb	170.8
Combustible per hour, lb	149.7
Water actually evaporated by boiler per sq. ft. of heating surface per hour	5.63
Water evaporated from and at 212 degrees F. per sq. ft. of heating surface per hour, lb	6.43
Water actually evaporated by boiler per sq. ft. of grate surface per hour, lb	187.8
Water evaporated from and at 212 degrees F. by boiler per sq. ft. of grate surface per hour, lb	214.0
Dry coal per sq. ft. of grate surface per hour, lb	22.8
Dry coal per sq. ft. of heating surface per hour, lb	0.68
Combustible per sq. ft. of grate surface per hour, lb	20.0

Combustible per sq. ft. of heating surface per hour, lb	0.6

Economic Results

Water actually evaporated by boiler alone per pound of dry coal, lb	8.25
Water actually evaporated by boiler alone per pound of combustible, lb	9.40
Equivalent evaporation from and at 212 degrees F., by boiler alone per lb. of dry coal, lb	9.4
Equivalent evaporation from and at 212 degrees F., by boiler alone per lb. of combustible, lb	10.72
Equivalent evaporation from and at 212 degrees F., by boiler and superheaters per lb. of dry coal, lb	10.72
Equivalent evaporation from and at 212 degrees F., by boiler and superheaters per lb. of combustible	12.22
Dry coal per boiler horse-power hour, boiler alone, lb	3.66
Dry coal per boiler horse-power hour, boiler and superheaters, lb	3.21
Combustible per boiler horse-power hour, boiler alone, lb	3.21
Combustible per boiler horse-power hour, boiler and superheaters, lb	2.82
Coal as fired per i.h.p. of engine per hour, lb	1.35
Dry coal per i.h.p. of engine per hour, lb	1.315
Coal as fired per b.h.p. of engine per hour, lb	1.39
Dry coal per b.h.p. of engine per hour, lb	1.36
Combustible per i.h.p. of engine per hour, lb	1.15
Combustible per b.h.p. of engine per hour, lb	1.19
Steam per i.h.p. per hour, lb	11.00

Steam per b.h.p. per hour, lb.	11.38
B.t.u. in coal supplied to engine per i.h.p. hour	17,820
B.t.u. in coal supplied to engine per b.h.p. hour	18,420

Efficiencies

Heating value per pound of dry coal B.t.u.	13,541
Heating value per pound of combustible B.t.u	15,500
Efficiency of boiler, furnace and grate, per cent	67.2
Efficiency of boiler, superheaters, furnace and grate, per cent	76.5
Thermal efficiency of engine based on i.h.p., per cent	17.05
Thermal efficiency of engine based on b.h.p., per cent	16.50
Thermal efficiency of entire unit, per cent	13.8
Efficiency of engine referred to the Rankine Cycle efficiency as unity	60.0
Mechanical efficiency of engine, per cent	96.8

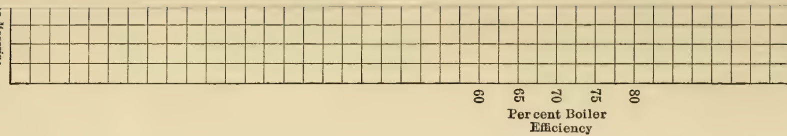
Boiler Heat Balance	*B.t.u.*	*Per cent*
Heat absorbed by boiler and superheaters per lb. of combustible	11,897.0	76.82
Loss due to moisture in coal per lb. of combustible	31.0	0.20
Loss due to hydrogen in the coal per lb. of combustible	420.0	2.72
Loss due to incomplete or CO per lb. of combustible	189.0	1.23
Loss due to chimney gases per lb. of combustible	1,920.0	12.28
Radiation and unaccounted losses	1,043 0	6.75
B.t.u. per lb. of combustible	15,500.0	....

The curves charted in Fig. 57, submitted by the Buckeye Engine Co., will serve to illustrate the relations of steam and fuel consumptions at various loads on a 150 horsepower Buckeyemobile.

Engineers have criticized as detrimental to the boiler the element of vibration communicated to it by the superimposed engine. As far as any published record of trouble goes there seems to be at present no evidence that the boiler is materially injured in the locomobile.

Unaflow Steam Engine

The unaflow engine invented by Professor J. Stumpf of Charlottenburg Hochschule marks a notable advance in the design of prime movers. As its name indicates, the steam from its entry to its exhaust flows in only one direction on either side of the piston. The idea of the invention is to produce the economy of a compound or triple-expansion steam engine by means of a simple single-cylinder engine, and this result has been achieved. The working of the engine involves a very early cut-off, a high compression, and a graduated distribution of temperature from admission to exhaust. The engine is primarily designed for high pres-



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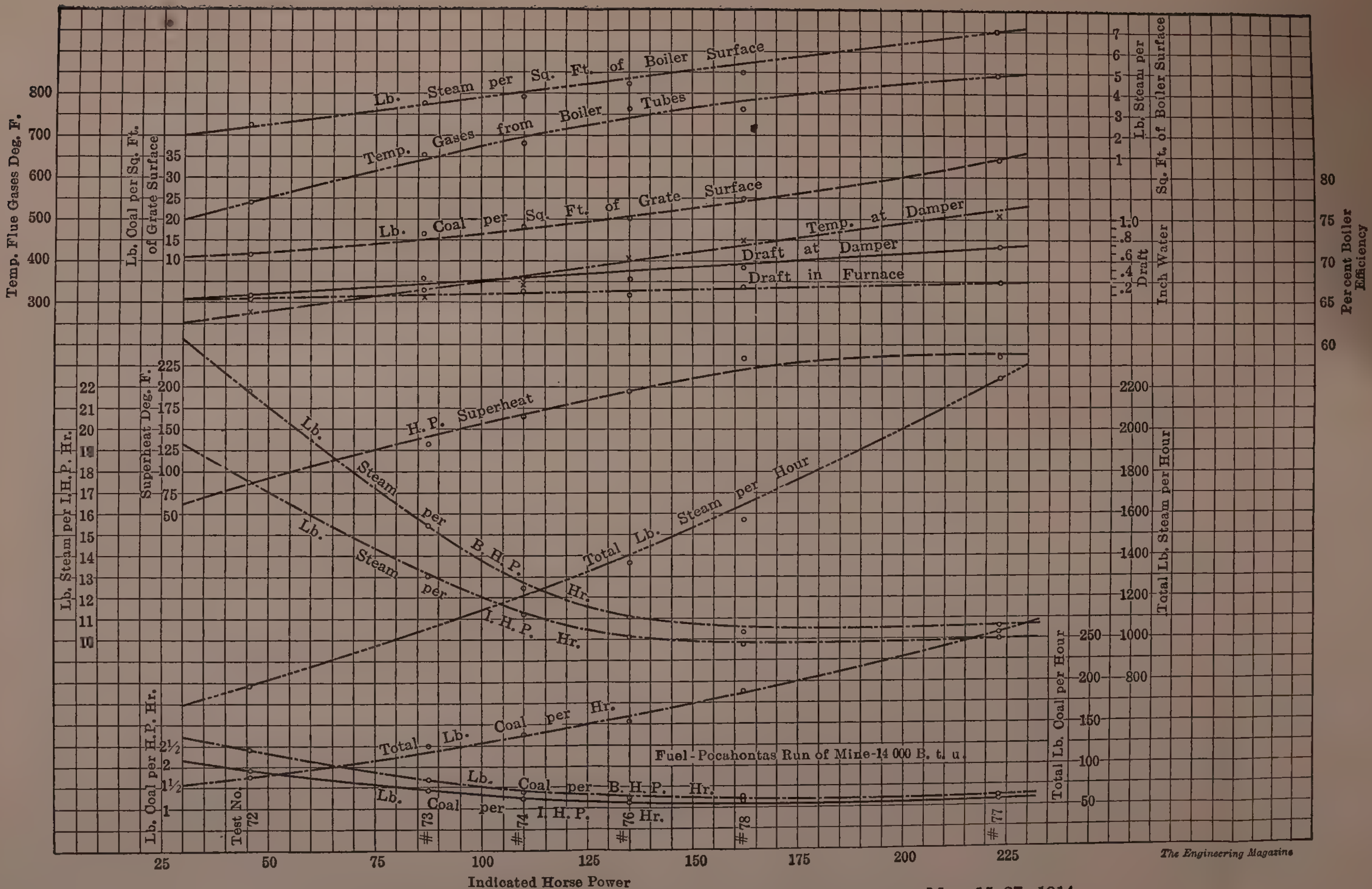


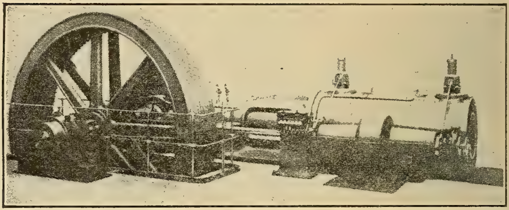
FIG. 57. ECONOMY TEST OF A 150 HORSE-POWER BUCKEYEMOBILE. MAY 15–27, 1914

sure and at the same time highly superheated steam, although remarkably high economies are obtained with saturated steam, in which latter case the steam jackets are extended over a greater portion of the cylinder.

As will be recalled from the discussion on steam-engine losses in Chapter VI, maximum economy is obtained by dividing the work of expansion of the steam into several stages or cylinders, thus developing the compound, the triple- and quadruple-expansion engines. Compounding is efficient only with high-pressure steam, which for complete expansion involves a wide range of temperature from the high point at admission to the low point at exhaust. If this steam were expanded all in one cylinder this cylinder would be subject to these wide variations of temperature. Thus at exhaust the cylinder would be cooled to a point approaching the temperature of the exhaust steam. Then when the high-pressure steam is admitted, owing to this wide difference of temperature, a heavy cylinder condensation would result. Consequently compounding was resorted to so that the steam entering each of the various cylinders would come in contact with metal approaching its own working temperature. In this manner the great cylinder condensation

which would occur in a single cylinder is greatly reduced.

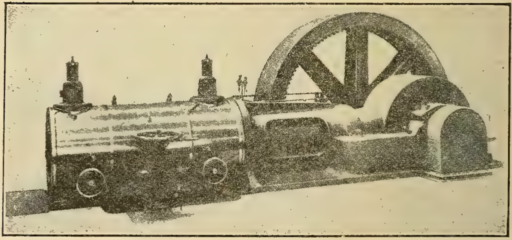
Professor Stumpf has produced a single-



Courtesy of *Steam*

Fig. 58. Sulzer Bros. Stumpf Unaflow Engine

cylinder engine, however, which, owing to its unaflow principle, maintains a graduated temperature in the cylinder walls and this

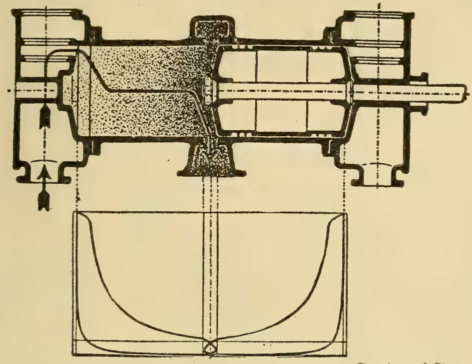


Courtesy of *Steam*

Fig. 59. Sulzer Bros. Stumpf Unaflow Engine

temperature approximates the temperature of the steam in the cylinder at all parts of the stroke, so that the condensation is re-

duced to zero in best practice and to a minimum in any case. Figs. 58 and 59 exhibit side views of a 500 horse-power Sulzer Brothers Stumpf engine. Fig. 60 shows diagrammatically the construction of the cylinder, piston and steam ports. It will be noted that the steam enters through the cylinder head

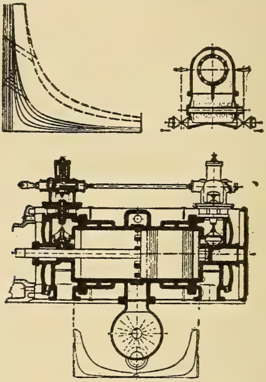


Courtesy of *Steam*

FIG. 60. STEAM FLOW IN SULZER BROS. STUMPF UNAFLOW ENGINE

and that the steam chest so formed acts as a jacket for the head of the cylinder. This maintains a maximum temperature at that part of the cylinder where high pressure and only high pressure is active. The path of the steam, including its admission, expansion, and exhaust is indicated by the arrow in this figure. The steam follows through the

stroke until the piston, which is of special hollow construction, opens the exhaust ports forming a belt surrounding the cylinder. Thus all exhaust valves are eliminated and consequently leakage from this source is reduced to a theoretical minimum.



Courtesy of *Steam*

FIG. 61. CONDENSER CONNECTION AND INDICATOR DIAGRAMS SULZER BROS. STUMPF UNAFLOW ENGINE

A glance at the arrow above mentioned explains the name of "unaflow." In the ordinary reciprocating engine the steam flows in two directions instead of one, and it exhausts at the same end of the cylinder from which it enters. Consequently the exhaust steam greatly reduces the temperature of that portion of the cylinder with which the

incoming hot steam will be in contact. This results in the great cylinder condensation, which frequently amounts to 25 per cent of the water consumption of the engine.

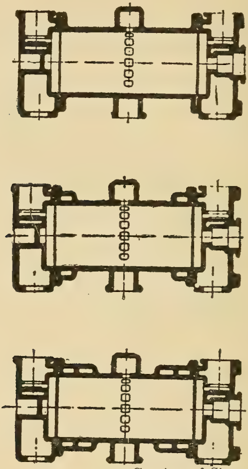
In the unaflow engine when exhaust has taken place the piston returns covering the exhaust ports and immediately compression begins. This compression takes place during about 90 per cent of the entire stroke of the engine. Consequently the compressed steam reaches a temperature about equal to that of the boiler pressure. This high compression, combined with the almost ideal admission and expansion of the steam, produces an indicator card which approaches the ideal Carnot cycle and explains the very high efficiencies obtained by the unaflow engine.

The piston is of special construction and is elongated so as to act as an exhaust valve in covering and uncovering the ports surrounding the middle section of the cylinder.

The clearance in this engine is reduced to a minimum because there are no exhaust ports at the head end as in ordinary engines, and this fact results in adding to the general efficiency of the design. The condenser is closely connected to the annular exhaust chamber as shown in Fig. 61, permitting a maximum vacuum to be effected in the cylinder.

When operating, if for any reason the vac-

uum breaks the compression produced will be above boiler pressure, since the terminal density of the steam will be suddenly increased. To take care of this emergency the inlet valve will lift from its seat to relieve the



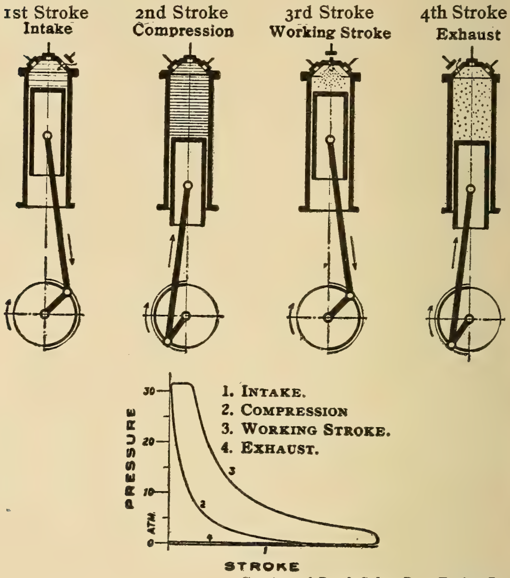
Courtesy of *Steam*

FIG. 62. STEAM JACKETING OF UNAFLOW ENGINE

pressure back to the boilers. When starting up, an additional clearance space may be opened into the cylinder by means of a hand valve in order to prevent excessive compression until full vacuum is established. This valve may also be made to operate automat-

ically in case of the breaking of the vacuum during operation. While the unaflow engine is primarily designed to take advantage of only high pressures and high superheat, it is also modified to produce very high efficiencies on saturated steam. This is accomplished by extending steam jackets (as shown in Fig. 62) so that they cover a greater surface of the cylinder in accordance with requirements. Thus in the top diagram of Fig. 63 only the cylinder heads are jacketed, and this design would be best adapted to highly superheated steam. The middle figure shows the steam jacket extending over a part of the cylinder beyond the heads, and would be adapted for a lower temperature steam; while in the bottom diagram the jackets extend still further to produce economical operation with saturated steam. The purpose in any of these cases is to prevent cylinder condensation, but in all three provision is maintained for keeping the high-pressure end of the cylinder hot and the low-pressure end cooler, so that the steam at any part of the stroke is always at a lower temperature than the surrounding walls of the cylinder, which prevents transferring of heat from the steam to the metal. The unaflow engine is capable of very great overload capacity and also produces flat economy curves. In fact,

the steam consumption per horse-power hour at wide variations of load varies less than with triple-expansion or compound engines.



Courtesy of *Busch-Sulzer Bros. Engine Co.*

FIG. 63

Professor Stumpf has applied his engine to locomobiles (described in another section of this chapter), with highly efficient results, and to locomotives which have been widely adopted in Germany, Austria, and Russia.

In marine practice it is claimed that one pound of coal has produced an indicator horse-power hour on 500 horse-power engines. In stationary practice these engines have been built in capacities as high as 8,000 to 10,000 horse power and are in successful operation. In this country we may expect a rapid growth in the utilization of the unaflow principle, and one firm at least is now engaged in the manufacture of this type of engine. The following test is taken from the literature of John Musgrave & Sons, Ltd., English manufacturers of the unaflow engine:

A test made on this engine by the Elsässischen Verein der Dampfkesselbesitzer (the Union of the Steam Boiler Users of Alsace), in February, 1909, gave the following results:—

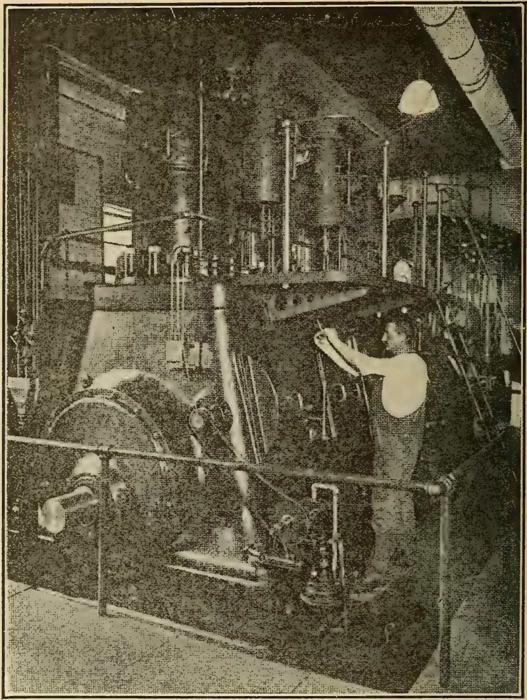
Power	494 i.h.p.
Cylinder	25¼ in. diameter
Stroke	3 ft. 3 3/8 in.
Speed	130 r.p.m.
Boiler pressure	178 lb. per sq. in.
Superheat	250 degrees F.
Vacuum in cylinder	26 in.

STEAM CONSUMPTION PER I.H.P. PER HOUR = 10.38 LB.

With 362 i.h.p. and 200 degrees F. superheat *the steam consumption was 10.55 lb. per i.h.p. per hour.*

Oil Engines

The Diesel and the De La Vergne represent the most efficient types of internal-combustion engines designed to operate on oil.



Courtesy of *The Busch-Sulzer Bros. Diesel Engine Co.*

FIG. 64. A DIESEL ENGINE

Both work on the four-cycle principle which is clearly illustrated by the diagrams of Fig. 63. Combustion occurs once in every four strokes or every two revolutions, thus:—

First stroke: intake of fresh air; piston traveling outward.

Second stroke: compression of this air; piston moving inward.

Third stroke: injection of fuel at dead centre, causing combustion and expansion of the

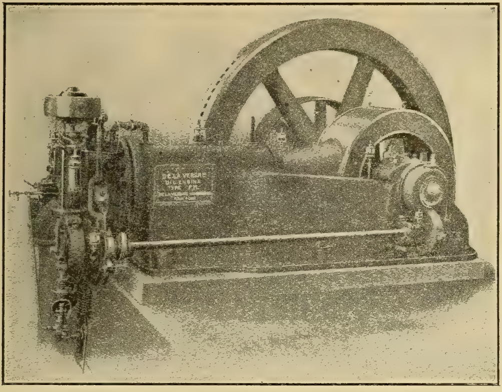


FIG. 65. DE LA VERGNE 90 HORSE-POWER OIL ENGINE

working stroke with piston moving outward.

Fourth stroke: expulsion of exhaust gases from cylinder; piston moving inward.

These events may be traced on the indicator diagram of the Diesel engine in the lower part of Fig. 63. Fig. 64 illustrates the Diesel engine and Figs. 65, 66 and 67 apply to the De La Vergne. In both of the

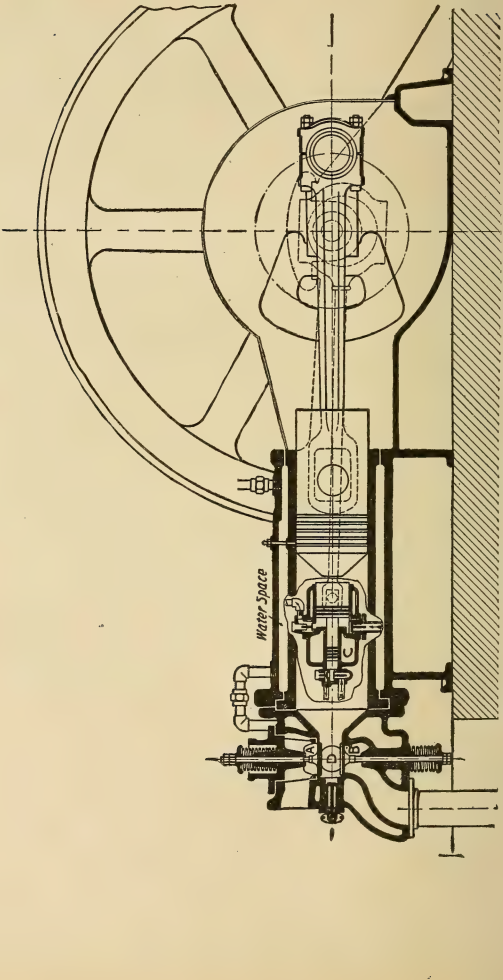


FIG. 66. LONGITUDINAL SECTION OF DE LA VERGNE OIL ENGINE

above engines the oil is atomized and injected into the cylinder by means of very highly compressed air. But the essential differences in their operation are as follows:

Compression. The Diesel compresses the fresh-air charge in the cylinder to 460 pounds per square inch; the De La Vergne to about 300 pounds.

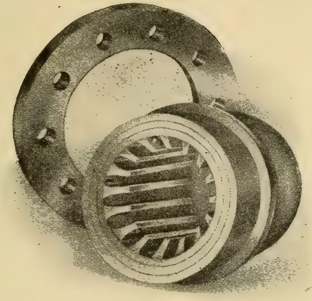


Fig. 67. De La Vergne Vaporizer

Ignition. The Diesel depends solely upon the temperature of the compression, about 1,000 degrees F., to ignite the charge of oil as it is sprayed into this heated air. The De La Vergne also makes use of its temperature due to compression, which is lower, but does not depend solely upon this agency. It employs in addition the use of a vaporizer which is a modified form of hot bulb. This

serves the double purpose of insuring ignition and of completing the gasification of the oil which is sprayed against its hot surfaces. This vaporizer is shown in detail in Fig. 67 and in its relation to the engine at D in Fig. 66.

INJECTION OF FUEL. In both engines the oil enters the cylinder by the action of air under high pressure but small in volume. The De La Vergne injects its oil instantaneously, in charges measured out by a small oil pump connected to the engine and timed with its action. The Diesel introduces its oil in a spray which continues over a considerable period of time, covering ordinarily one-tenth to one-eighth of the stroke of the piston.

GOVERNING. In the De La Vergne the governor controls by a cam mechanism the length of stroke of the oil-supply pump and thereby causes it to measure out to the engine the exact amount per cycle which may be required to maintain the regular speed of the engine for all variations of the load.

The Diesel governor regulates the time period of injection according to load requirements. Thus the oil will continue to enter the cylinder as the piston moves forward from dead centre as far as say 12 per cent of the stroke, or to such point as may be required to meet the load conditions. Thus this ef-

fect of fuel admission may be compared to the steam admission on a Corliss engine, and this together with the automatic and quick termination of the admission period results in an indicator diagram which strongly resembles that produced by this type of steam engine.

From the standpoint of *thermal efficiency alone,* highly developed oil engines of the above types are superior to any other class of prime movers. Their *commercial efficiency* varies with, and depends upon, the cost and reliability of oil supply, the size and cost of plant, the character of service, and other determining values dependent upon local conditions.

With the Diesel engine efficiencies as high as 32 to 35 per cent based on net useful output are claimed. With an oil of 19,000 B.t.u. per pound this would mean 0.42 pounds of fuel per brake horse-power hour. The characteristically "flat" economy curve of this engine is shown in Fig. 68. Efficiencies equal to these both at full and at partial loads are claimed for the De La Vergne engine.

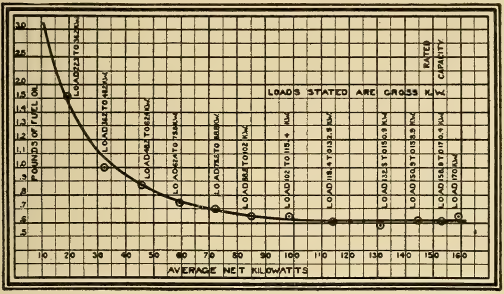
One of the greatest advantages of an oil-engine plant is that the prime mover alone constitutes an entire plant in itself. Gas producers or boilers are eliminated, and attend-

ance is reduced except in very small undertakings, and the factor of reliability of the oil engine approaches that of the steam plant. As compared to the gas engine in this respect the oil engine has two distinct advantages. The feeding of its fuel in accurately measured amounts is positive, and for ignition of the charge it is not dependent upon electrical complications as is the gas engine. The reliability of the oil engine is being demonstrated by the rapid increase of its application for marine use, where an important saving of space and weight is also accomplished.

The fuel cost of power with an oil engine depends upon the efficiency, the load factor and the price and B.t.u. of the available oil supply. Conditions representing good factory practice would combine load factor, efficiency of engine and B.t.u. of oil such as to produce an average brake horse-power hour for about 0.6 pounds of oil. Assuming the cost of the oil to be \$0.025 per gallon containing $7\frac{1}{2}$ pounds, the fuel cost per brake horse-power hour will be $0.025 \div 7.5 = \$0.00333 = 3\frac{1}{3}$ mills. If the oil cost \$0.05 per gallon the fuel cost per brake horse power will be doubled.

For marine purposes on a small scale a special carburetor has been devised which permits the use of kerosene in regular gaso-

line engines, thus making available a cheaper fuel. This device has also been applied to automobile engines. The idea involved is the gasification of the less volatile fuel by the application of heat furnished by the exhaust



Courtesy of *Busch-Sulzer Bros. Diesel Engine Co.*

FIG. 68. DIESEL FUEL CONSUMPTION PER KILOWATT HOUR AT VARIOUS LOADS

gases from the engine. The gas is then used in the engine in the ordinary manner. The troublesome electric ignition system is not eliminated, and the use of this style of oil engine has received little if any encouragement for factory power-plant application.

Chapter XIX

REPORTS

AN owner's decision to have an investigation made in his plant is frequently brought about by one or more problems of specific nature which require immediate solution. The manner in which such a case is handled will often determine the whole operating economy of the plant during many years to come, either for better or for worse; and in the long run will involve either continuous items of efficiency and satisfaction or of waste and expense.

I have chosen a case of this class by way of supplying definite answers to some of the concrete questions which our previous more or less abstract discussions may have provoked. Owing to space limitations these reports have been greatly abbreviated. But since the second contains a comparative record of operating economy after the recommendations of the first report had been car-

ried out, it will be of particular interest, and more especially so because the questions treated are among those which confront many manufacturing plants at the present time. Probably no one case of investigation and improvement can be selected as typical, either as to saving effected or problems involved. The present instance contains a sufficient variety of questions, however, to claim at least the title of ''average'' from that point of view.

The boiler plant was found to be highly efficient, an unusual occurrence in the course of my work; and cheap fuel was being used so that but little saving could be made at that point. Therefore, since practically all the economies had to be found in the remaining departments, the saving of $5,000 a year, based on operation during the acceptance tests, may be considered good under the circumstances, especially in a plant of only 400 boiler horse-power rating with small running expenses.

The mill owners were under pressing need of more power owing to a rapid growth of their business. They had a good compound condensing engine supplied by two 6½ by 20-foot horizontal tubular boilers at 125-pounds pressure, with usual condenser, feed, and fire-pump equipment. Besides the steam

power, they were purchasing additional electric current at 2½ cents per kilowatt hour. The mill was heated throughout, including a supply for process work, with live steam from the boiler reduced to 20-pounds pressure. The system comprised both direct and indirect heating.

The management was progressive and efficient. This briefly comprised the situation when I was called into consultation. The problem in hand resolved itself into three specific questions which may be stated as follows:—

1. Should all future increase of power be provided by the purchase of outside electric current, leaving the steam plant to furnish its present amount of power?

2. Should the use of the present steam-power plant be discontinued entirely, and all the machinery of the mill be driven by purchased current? In this event the power company offered a rate of two cents per kilowatt hour.

3. Should the mill throw out all purchased current and make its own power exclusively, increasing its plant equipment for this purpose? If so, what style of plant would best meet these conditions?

In order to determine these questions intelligently, it was necessary first to know at

what cost, *all charges included,* the present steam plant was making power. This could not be learned without an investigation as to the amount of live steam used for heating, since this item would directly determine that portion of the boiler-plant overhead and labor expenses which would be chargeable exclusively to heating. Special tests for this purpose were planned and conducted.

It was also desired to learn at what cost per kilowatt hour all the power *could* be made with added and improved steam equipment.

Since a large portion of this item would be inversely proportional to the *amount* of power required during the year, special tests were made in order to obtain the twenty-four-hour load curves both of the present engine and of the power furnished by the purchased current.

The results of all the tests that were made both on the steam and electric power were carefully checked up with the company's records of cost for fuel, labor, repairs, interest charges, and expenditures for the purchased current. This is a most essential procedure in all investigating work, in order that the total power output per year may be accurately obtained to insure substantial correspondence of factory records with the actual test

results, and to obtain a mathematically correct figure on the increase in steam requirements during the winter months. This work involves a thorough acquaintance with the methods employed by the book-keeping department of the business and a most careful analysis of the data on record. It has to be remembered that the head book-keeper does not pretend to be an engineer, and his data usually require the most painstaking analysis by the investigator in order to divide the expenses into the particular items applicable to the work in hand.

In this instance the available records were unusually complete and clear, and it was possible to save much time in this part of the work. There was no difficulty in enlisting the interest and help of the power-plant operating force, and this fact proved of great value in arranging for and conducting the various tests which are recorded.

For convenience to the executives of a business I set down my recommendations and findings in a condensed form at the head of my report, so that the gist of the matter may be found within a few pages. The records of all tests and the complete data to substantiate these conclusions are then given in full and properly indexed for quick reference. Other matters than those outlined

were also taken up in these reports, but many of them are omitted for the sake of simplification and of direct dealing with the principal problems involved.

A little story connected with these reports which are to follow does not appear within their pages. Before I was called in, one of the large electric manufacturing companies was requested to send an engineer to make an examination of the power situation at the mill for the purpose of reporting recommendations to suit the case in hand.

This they did, and their report called for an expenditure for steam and electric apparatus, which they specified, amounting to $40,000.

My report for accomplishing the desired result, which was put into effect, called for an outlay of only $7,000, including installation, with this difference: my plan provided for the utilization of exhaust steam; their plan did not. In the execution of the adopted plan the electrical machinery was purchased from this same company, but my client saved $33,000 in first cost. Many morals might be drawn from this little incident, but their deduction may as well be left to the reader.

As far as possible in these reports I have omitted those sections of each which appear to be treated in the other to a sufficient ex-

tent to inform the reader, at least by implication, of all the important features under examination.

The net dividend on the investment of $7,000 was found to be over $5,000 per year, as shown in the second report, thus producing returns to my clients of 71½ per cent per annum on their outlay of capital.

INVESTIGATION AND REPORT ON STEAM AND FUEL CONDITIONS AND COST OF POWER

DIVISIONS OF THE REPORT

OBJECT AND METHOD OF INVESTIGATION

The principal object of the tests made at this plant was to determine the following questions:—

1. Would it be economical to discontinue the present steam-power plant and buy all the power required to run the mill, at the rate of 2 cents per kilowatt-hour?
2. Would it pay to continue the present steam plant for present requirements (as now carried by said plant) and depend upon purchased electric power for all additional requirements and for increase of power at 2½ cents per kilowatt hour?
3. Would it pay for this plant to discontinue buying electric power and enlarge the present plant to make all the power required; and if so, what kind of equipment would best suit the local conditions for highest commercial efficiency?

The following tests furnished basic information in the investigation and determination of these questions:—

a. Day and night loads on the boiler plant.
b. Day and night loads on the engine plant.
c. Day and night loads on the purchased electric power.
d. The efficiency of boiler plant and engine plant, both separate and combined.
e. The amount of steam or boiler horse power required for heating and mill-process work as compared to the steam used in the engine.
f. The costs of evaporation both with the usual mixture of dust and slack, and with slack alone.
g. Comparison of electric *versus* shafting and belting drive from present engine, etc.

The method of testing was to take conditions exactly as found, the engineer and fireman being

asked to continue their duties in every way exactly as though the writer was not on the premises. A constant effort was made to maintain these actual *working* and *not test* conditions.

CONCLUSION

The final result of this investigation has shown in a very decided manner that the most economical solution is for this plant to make all its own power and discontinue buying electric power.

Present Plant.—The present engine plant is making power at the rate of $0.0132 per kilowatt per hour, even when the engine is running below its normal capacity. If the engine is speeded up 10 per cent and run at an average of 300 horse power, a kilowatt hour would be produced for $0.00971.

Cost per Kilowatt Hour.—These costs include all charges against power, such as depreciation, interest, operating and fuel cost, all having been based on actual tests and data from the company's books.

Additional Power.—It has been shown by special tests and computation submitted herewith that this plant can install a new electric generator set and make all the electric power now purchased (at 2½ cents) for $0.01238 per kilowatt hour with the present load, and for about $0.009 when this load reaches 1,000 kilowatt hours per day. This is based upon the utilization of one-half of the exhaust steam from the new unit, a safe figure borne out by my tests.

Saving.—Actually, however, the utilization of exhaust steam will reduce the *fuel* per kilowatt hour to *less* than one-half, so that the above costs may be considered as high figures. There will be a saving of at least 1.26 cents per kilowatt hour at the present light load, which amounts to $7.34 a day

when using the present amount, i. e., 582 kilowatt hours a day (average working days, 1908).[1]

There will be a charge of 50 cents a day for depreciation and interest on new exhaust apparatus, making the net saving $6.84 a day or $2,052 a year of 300 working days, with the present light load.

The addition of the unit recommended will increase the present engine-plant capacity about 50 per cent.

Future Saving.—With the rapid growth of the plant now predicted it is safe to state that about 1,000 kilowatt hours per day will soon be required instead of the 582 kilowatt hours now purchased from the A. & S. Power Co. When this occurs, your new unit will be making a kilowatt hour for about $0.009 as above stated, thus saving (0.025–0.009) = 0.016 per kilowatt hour over the A. & S. Power Co.'s rates. This is $16 per day or $4,800 per year. Or if the interest on heating apparatus be deducted, the net saving will be $4,650 per year.

The saving will be still greater as time goes on and the load increases.

As an investment, based on the estimated saving a year hence, this company would receive 4,650 ÷ 8,000 = 58 per cent net returns, all charges taken into account.

RECOMMENDATIONS

Continue Present Engine.—Continue the present steam plant and increase the available power by speeding engine up 10 per cent, i. e., to 95 revolutions per minute.

Overhaul Engine.—Have the engine thoroughly overhauled and tuned up to reduce the steam consumption.

[1] This power consumption increased to 1,070 kilowatt hours, thus showing a correspondingly greater saving as per second investigation after recommendations were carried out.

New Unit Recommended.—Install a direct-connected simple Corliss engine and generator set for making the necessary electric power instead of buying this power, this generating set to be run at night and usually in the day time except when the load on the main engine is running very light. Make this new set of 90-kilowatts rated capacity. It would then run at the present load at about one-third of its rated power, which would give an extra 60 kilowatts of available power for increasing the factory load.

This unit should be of as slow speed as possible i. e., in the neighborhood of 125 to 130 revolutions.

Utilizing Exhaust.—It is recommended that the exhaust steam from this unit be used in the present heating fan and in the contemplated new heating fan for heating all parts of the mill, and also in a new and larger feed-water heater for raising the temperature of the boiler water.

Vacuum System.—A vacuum return-line system is required to give proper circulation of the exhaust steam in the fan coils. By extending this vacuum system to the office radiators [1] this engine exhaust could be used in heating the office as well.

Night Man.—It is recommended that a good man be put on at night when the new engine is installed to look after same, and also to take charge of the boilers. Such a man would probably save 10 to 20 per cent of coal for the same production of steam as at present.

Bank One Boiler at Night.—With the small amount of steam used at night at the present time the boilers are run at only about 17 per cent of their rated power. Even if it is not thought advisable to shut off the steam from one boiler at

[1] Both direct and indirect heating were employed.

night, one fire should be banked, and the damper, fire-doors and blower tubes kept tightly closed. Thus by making a single boiler do the whole work the rate of combustion would be doubled, a higher fire-temperature would be maintained, and less superfluous air would get through the grates to cool the boilers. This would result in a substantial saving.

Separator and Receiver.—The new engine should have its steam pipe fitted with a combination receiver and separator to supply the engine with dry steam and a uniform pressure. It would also tend to prevent vibration of the steam line.

Steam for Mill Requirements

THREE TESTS

TEST NO. 1

Rough Preliminary Test

Evaporation of No. 1 boiler when supplying mill requirements alone, main engine being cut out. Test made in afternoon.

Date	Oct. 14, 1909
Duration, hours	4
Pounds evaporated	12,200
Pounds evaporated per hour	3,050
Steam gáge, pounds	120
Temperature of feed water, degrees F	161
Factor of evaporation	1.0969
Equivalent evaporation per hour	3,353
Boiler horse power developed	97.2
Rated horse power of boiler at 12 sq. ft.	207
Percentage of boiler horse power developed	47

TEST NO. 2

Second test on No. 1 boiler when supplying mill requirements only.

Date	Oct. 15, 1909
Duration, hours	8
Pounds evaporated	22,050
Pounds evaporated per hour	2,756
Steam gage, pounds	110
Temperature of feed water, degrees F.	161
Factor of evaporation	1.0951
Equivalent evaporation per hour	3,017
Boiler horse power developed	87.5
Rated horse power of boiler	207
Percentage of boiler horse power developed	42.3

In this test No. 1 boiler was shut off from the engine by the usual valves for the purpose but was not blanked off with flanges. Outdoor temperature at 8 a.m. was 46 degrees. At 3 p.m. was 53 degrees. The heating load during this test was as follows: Four felt dryers were on all day; part of the office steam on. Two coil dryers were on, but scouring room and big fan were off during the test.

TEST NO. 3

Test No. 3 on steam used in mill line, *steam line blanked* off to mill only.

Date	Oct. 18, 1909
Duration of test (*9.35 a.m.—5.05 p.m.*) hours	7½
Average steam pressure, pounds	110
Average temperature of feed water, degrees F	158
Factor of evaporation	1.0982
Total water evaporated, pounds	42,000
Water evaporated per hour, pounds	5,600

Water evaporated per hour from and at 212	6,150
Average boiler horse power developed	178.3
Maximum boiler horse power developed (1 hour)	229
Minimum boiler horse power developed (1 hour)	140
Rated horse power of boiler	207
Percentage rated horse power developed (average)	86.2

On this day 4,000 pounds of wool were scoured, (on some days 6,000 pounds are scoured) and all heating apparatus was on (except part of the office steam) including the large fan and the scouring room.

During this test an estimate of the horse power was made each hour and found to be approximately as follows:

	Horse Power to Mill
9.35—10.35	140
10.35—11.35	229
11.35—12.35	178
12.35— 1.35	152
1.35— 2.35	197
2.35— 3.35	190
3.35— 4.35	165
4.35— 5.05	178

The weather was warm, with rain all the afternoon.

STEAM USED AT NIGHT

A test of thirteen-hours duration was made from 6 p.m. to 7 a.m. to determine the amount of steam made at night which is entirely consumed for heating purposes.

Both boilers were run as usual and fired by the night-watchman.

Results of Test

Date	Nov. 3 & 4, 1909
Duration, hours	13
Water evaporated, pounds	28,764
Water evaporated per hour, pounds.	2,213
Temperature of feed water, degrees F	154
Steam gage, pounds.	105
Water evaporated from and at 212 degrees, pounds	2,434
Coal of usual mixture consumed, approximately, pounds	4,850
Water per pound of coal—approximately, pounds	5.93
Water per pound of coal from and at 212 degrees (f = 1.1004) pounds.	6.52
Average boiler horse power developed	*70.6*
Rated horse power of boilers	414
Per cent rated horse power developed	17.2

Owing to running the boilers at only 17.2 per cent of their normal rated capacity the consumption of fuel is way out of proportion to the production of steam. If a single boiler could carry the night load the combustion would be improved and less coal would be burned.

To check up the results two tests on the flue gases were made which showed 3 per cent to 3½ per cent of CO_2. This simply means that several hundred per cent excess air is admitted to the fire owing to the large percentage of unused or dead grate surface.

This test does not show what the maximum night heating requirements may be, as the outside temperature was only 43 degrees at midnight, and the large fan was run only 7½ hours out of the 13 hours of the test. The data and results of tests on No. 1 boiler are given on pages 523 to 529. For convenience of typographical arrangement the flue-gas analysis precedes.

DATA AND RESULTS OF TWO EVAPORATIVE TESTS ON NO. 1 BOILER

FLUE-GAS ANALYSES

For Fuel Data see page 524	*Mixture Test*	*Slack Test*
Average interval at which coal was fired, minutes	13.7	11.7

Mixture Test	CO_2	O	CO
10.07 draft off	10.50	....	....
10.50 " "	8.90	....	....
11.45 " on	15.50	....	....
12.47 " "	14.70	....	....
2.33 " "	15.30	....	....
3.03 " "	9.00	....	....
3.52 " "	11.20	....	....
4.45 " off	10.30	....	....
4.55 " "	9.30	....	....
Average CO_2—11.63			
Slack Test			
10.35 draft on (dense smoke)	13.7	3.3	1
11.36 draft just off (dense smoke)	7.2	11.8	0.3
1.10 draft off	4.5	....	....

DATA AND RESULTS OF TWO EVAPORATIVE TESTS ON NO. 1 BOILER TO DETERMINE EFFICIENCY AND COST OF EVAPORATION, AND STEAM BLOWER CONSUMPTION

	Test No. 1	*Test No. 2*
Kind of fuel	By weight, 5 parts anthracite screenings, 1 part bituminous slack	Slack, bituminous See note below
Kind of furnace	6 ft. 3 in. × 6 ft. grate, 34 in. from boiler shell	
Kind of boiler	H. T. 6 ft. 6 in. by 20 ft. 128 3½-in. tubes	
Method of starting and stopping test	Alternate	
Date of trial	10/19/09	10/21/09
Duration of trial	8 hours	8 hours

The test on bituminous slack coal was made simply to demonstrate its results under present conditions of equipment and operation. The regular fuel used was five parts anthracite screenings and one part bituminous slack.

Dimensions and Proportions		
Grate surface, sq. ft.	37½	
Approximate width of air spaces in grate	about 5/16 in.	
Proportion of air space to whole grate surface	37½ per cent	
Water-heating surface	2,484 sq. ft.	
Superheating surface		
Ratio of water-heating surface to grate surface	66.25 to 1	
Steam pressure by gauge	115	118
Force of draft between damper and boiler	0.20 in. to 0.25 in.	0.05 in. to 0.10 in.
Draft pressure under grate, inches water	0 to av. press. of 0.73 in.	0.10 in. to 0.40 in.
Draft pressure over grate, when blowers on	positive	positive
Average Temperatures		
Of feed water entering boiler	159 degrees F.	160 degrees F.
Of escaping gases from boiler	435 degrees F.	448 degrees F.
Of air entering ashpit	60 degrees F.	60 degrees F.
Fuel		
Weight of coal as fired	3,344 lb. (hard—2,787, soft—557)	3,955 lb. soft
Percentage of moisture in coal	hard, 5 per cent soft, 4.6 per cent	3.43 per cent

DATA AND RESULTS OF TWO EVAPORATIVE TESTS ON NO. 1 BOILER, ETC.—*Continued*

	Test No. 1	*Test No. 2*
Fuel—Continued		
Total weight of dry coal consumed..........	3,179 lb.	3,819 lb.
Total ash and refuse dropped through and cleaned from grate, lb....................	529 lb.	
Percentage of ash in dry coal based on chemical analysis..........................	16.7 per cent	
Fuel per Hour		
Coal as fired consumed per hour.............	418 lb.	494 lb.
Dry coal consumed per hour................	397 lb.	477 lb.
Dry coal per sq. ft. of grate surface per hr...	10.6 lb.	12.7 lb.
Calorific Value of Fuel		
Calorific value by oxygen calorimeter, per lb. of dry coal..........................	Anthracite, 12,800 B.t.u.; bituminous, 12,188 B.t.u.	13,167 B.t.u.

Moisture in coal as delivered to fireman	Anthracite, 5 per cent.; bituminous, 4.6 per cent	3.43 per cent
Water		
Water actually evaporated	26,951 lb.	30,453 lb.
Factor of evaporation	1.0981	1.0961
Equivalent water evaporated from and at 212 degrees	29,592 lb.	33,376 lb.
Equivalent water evaporated, corrected for steam used by the blower	29,490 lb.	33,256 lb.
Water per Hour		
Actual water evaporated per hour	3,369 lb.	3,807 lb.
Equivalent evaporation per hour per sq. ft. of heating surface	1.487 lb.	1.678 lb.
Equivalent evaporation per hour used by blower	12.75 lb.	15.0 lb.
Per cent of time blower was on, about	34 per cent	40 per cent
Equivalent evaporation per hour from and at 212 degrees	3,699 lb.	4,172 lb.

DATA AND RESULTS OF TWO EVAPORATIVE TESTS ON NO. 1 BOILER, ETC.—*Continued*

	Test No. 1	*Test No. 2*
Horse Power		
Total horse power developed	107.2	121.0
Horse power developed, corrected	106.8	120.5
(For steam used by the blowers)		
Builders' rated horse power at 12 sq. ft. per horse power	207	207
Percentage of rated horse power developed, useful steam (106.8 ÷ 207) (120.5 ÷ 207)	51.6 per cent	58.2 per cent
Percentage of horse power used by the blowers (107.2 ÷ 0.405)	0.38 per cent	0.42 per cent
Economic Results		
Water apparently evaporated under actual conditions per lb. of coal as fired	8.06 lb.	7.71 lb.
Equivalent evaporation from and at 212 degrees per lb. of coal as fired, not corrected for blower consumption	8.86 lb.	8.44 lb.

Equivalent evaporation per lb. of coal as fired, corrected for steam consumption of blower.	8.825 lb.	8.42 lb.
Efficiency		
Efficiency of boiler, including grate, based on the available heat in the coal as fired......	71.1 per cent	64.2 per cent
Available heat in one lb. of mixed coal as fired, after deducting loss due to moisture and heat to evaporate the moisture............	12,011 B.t.u.	12,673 B.t.u.
(Anthracite, 12,099 B.t.u.; bituminous, 11,571 B.t.u.)		
Cost of Evaporation		
Cost of coal delivered and unloaded, bituminous slack \$2.40 + 0.27 for 2,240 lb. Per 2,000 lb..............................	\$2.48	\$2.48
Anthracite dust or screenings for 2,000 lb..	\$1.90	
Cost of coal for evaporating 1,000 lb. of water from and at 212 degrees..................	\$0.1133	\$0.1473

NOTE: 1 ton (2,000) lb. of mixture of dust and slack costs \$2.00 at fire room. If the bituminous coal were burned at an efficiency equal to the mixture test the cost of evaporation with the bituminous coal would be \$0.133.

BOILER-PLANT CAPACITY

The normal rating of each of the horizontal tubular boilers is 207 boiler horse power. This rating provides 12 square feet of water-heating surface per horse power. (Two boilers = 414 rated horse power.)

A boiler horse power means the evaporation of 34½ pounds of water into steam at atmospheric pressure from a feed-water temperature of 212 degrees F.

With the prevailing conditions as to temperature of feed water and boiler pressure at this plant, the factor of evaporation is 1.097, so that it takes 31.5 pounds of water actual weight at this feed temperaature to produce a boiler horse power. (That is, $31.5 \times 1.097 = 34.5$.)

It is good practice to force boilers of this type 50 per cent above normal rating, so that (150 per cent $\times$ 414 horse power) = 622 boiler horse power may be had if required.

The engine consumed 19.32 pounds of steam (water) per horse power per hour, whereas in this plant a boiler horse power is 31.5 pounds of water. Therefore under these conditions an engine horse power is $19.32 \div 31.5 = 61.4$ per cent of a boiler horse power.

If therefore the engine were speeded up and were to operate at 300 horse power continuously, it would require only 61.4 per cent of 300 = 184.2 boiler horse power. This would leave for process work, heating, or additional power 622 — 184 = 438 boiler horse power. Now actual tests on steam required at this time (October and November) of year for heating and mill processes showed 70 boiler horse power at night and from 87.5 boiler horse power and 97.2 boiler horse power to 178.3 boiler horse power average loads, and for a single hour ran up to 229 boiler horse power.

Suppose in the winter time the mill requirement should be just double the highest average obtained

in the highest test, i.e., $2 \times 178 = 356.6$ boiler horse power. Suppose at the same time the engine is taking its maximum of 184.2 boiler horse power, then the total extreme maximum requirements would be $184.2 + 356.6 = 540.8$ boiler horse power. Even under these conditions there would still be $622 - 541 = 81$ boiler horse power available for any purpose desired.

BOILER-PLANT EFFICIENCY

During the day run, when a greater proportion of coal is burned, this plant is operating at very high efficiency—considerably higher, in fact, than the average factory boiler-plant. This high efficiency may be attributed principally to the combination of small grate-surface and good grade of cheap anthracite, mixed with enough slack to make it free-burning, and to good handling of the fire, together with the system of forced draft in use.

Average boiler-plant efficiency will not as a rule run over from 61 to 64 or 65 per cent, and in many cases will be found as low as 55 per cent, and sometimes as low as 50 per cent.

Boiler efficiency simply means that percentage of the available heat in the coal which is converted into useful steam. At this plant this efficiency in a day run was determined under regular working conditions, and was found to be $71^1/_{10}$ per cent. This is unusually good when it is considered that the plant was operating at only a little over one-half its normal rated capacity. That is, in the two efficiency tests made one boiler developed 106.8 horse power and 120.5 horse power as compared to its normal rating of 207 horse power.

A rough test was made to determine the economy of the present system of blowers for producing forced draft. These tests would give only approximate results, but at the same time would be a close

enough check to show whether any great waste is going on at this point. The writer has made tests on three different steam-induced draft appliances, these tests all being made in the same manner, and the system found here has given apparently better results than either of the other two referred to.

The amount of steam consumed in the blower when it is on at its full capacity, that is, two nozzles of $^3/_{16}$ inch diameter, would be in the neighborhood of 6 boiler horse power per hour for each blower. This would be 12 horse power for two nozzles which constitute a blower for one boiler. Under these conditions a single boiler would be developing probably in the neighborhood of 400 horse power, so that the percentage of steam used by the blower consisting of two nozzles would be 3 per cent of the boiler output.

Additional tests were made during the efficiency tests. The steam from a nozzle connected up similarly to the nozzles of the draft appliance was inserted in a barrel of water and the condensation was weighed. The end of the nozzle was inserted the same distance below the surface of the water as there were inches of draft pressure produced in the ash-pit. The result of this test gave approximately 0.4 of a boiler horse power per hour, which divided by 107.2, the horse power developed by the boiler, gives a consumption of 0.38 of 1 per cent as the amount of steam used by the boiler when the latter is run at about one-half its capacity. This percentage would increase when the draft is on a greater part of the time.

The flue-gas analyses made during the efficiency test gave an excellent check on the results obtained. These gas analyses show a very high degree of combustion and an excellent regulation of the air supply under the light-load conditions. The amount of grate surface used at this plant is very small compared with what is usually found for burning poor

grades of anthracite. The usual ratio of water-heating surface to grate surface would be about 40 to 1, whereas in this plant the ratio is 66.25 to 1. This is good practice as long as sufficient capacity can be obtained, but it is quite possible that under future conditions it will be necessary to enlarge this grate surface by increasing the length. This can be regulated according to requirements. (This change was made later on when more steam was needed.)

An efficiency test was also made burning slack bituminous coal. The result was only 64.2 per cent efficiency as compared to 71.1 per cent with the anthracite coal. The cost of evaporation with the bituminous coal alone was $0.1473, as compared to $0.1133 for the dust. (The grates and furnace are not adapted to soft coal.)

The actual calorific values of these two coals were: mixture five parts dust to one part slack, 12,011 B.t.u. available heat, and for the slack as found in the test, 12,673 B.t.u. available heat. Now a ton of 2,000 pounds of the mixture costs $2.00 at the fire-room, whereas the bituminous costs $2.48 for 2,000 pounds. If the bituminous coal were burned at an efficiency equal to that obtained in the mixture test, the cost of evaporation with the bituminous coal would still be 13.3 cents as compared to 11.3 cents for the mixture of dust as tested.

In conclusion the writer would state that this plant has unusual value in the quality and price of coal as used. Also the thermal efficiency of the plant is high owing to the equipment and operation, as above described. Combination of these two factors results in very low cost for evaporation. For purpose of comparison, a plant in Worcester which has improved combustion apparatus and is running at a higher thermal efficiency than this plant has to pay $3.75 for coal, and the consequent cost for evaporation runs about 15 cents per thousand pounds as compared to 11.3 cents at this plant.

POWER-PLANT OPERATING COSTS

(Not Including Purchased Power)

For the Year 1908

A—Total Light, Heat and Power Acct.............	$2,237.17	
Including Labor (engineer, day; fireman, day and night) Labor of repairs Labor of maintenance Labor of maintenance of motors (24 days each month)		Aug. 1909 $197.39 Sept. 1909 $175.62
B—Supplies.................	$4,921.05	
Including oil, waste, fuel ($3,201, dust; $911, slack; $4,112, fuel total), repair supplies, etc. (24 days each month)		Aug. 1909 $465.80 Sept. 1909 $365.21

Days of actual operation in 1908, 263.

Average day cost for 1908, actual operating days, $27.60.

Total operating cost 1908, $7,158.

The fuel therefore costs 4,112 ÷ 7,158 = 57.3 per cent of total operating costs. This includes heating.

15 per cent depreciation and interest on $20,000 plant = $3,000 year to add to the above costs.

Including depreciation and interest, the fuel cost is 4,112 ÷ 10,158 = 40.5 per cent of total cost of operating plant for heat and power.

COAL USED, AUGUST, 1909

	Dust	*Slack*
August 1909....	120 tons—$229.56	24 tons—$57.70
Sept. 1909......	125 tons—$237.50	25 tons—$60.00

PURCHASED CURRENT, YEAR 1908

C—Light.............................. $209.74
August 1909—$11.87
Sept. 1909 — 16.01

Power.............................. 3,819.84
August 1909—$304.08
Sept. 1909 — 275.80

Average day cost purchased, based on actual days operation—

Power.....................$14.52
Light..................... 0.80

COST OF POWER WITH PRESENT PLANT

From a boiler test made for the purpose it was found that a 5 to 1 (by weight) mixture of anthracite dust and slack costing $2.00 a short ton at the fire-room, would give under working conditions an evaporation of 8.06 pounds less 0.38 per cent used by blower = 8.03 pounds water evaporated per pound of coal under actual conditions.

From an all-day engine test under actual working conditions it was found that 19.32 pounds of water (steam) was required per indicated horse power per hour. Therefore one indicated horse-power hour consumes 19.32 ÷ 8.03 = 2.407 pounds of $2.00 coal. Hence the coal cost of producing one indicated horse-

power hour is (2.407 ÷ 2,000) × 2.00 = \$0.002407. Adding 10 per cent to this figure for stand-by losses gives:

I—Fuel cost per indicated horse power per hour \$0.002648

DEPRECIATION AND INTEREST CHARGES ON THE STEAM PLANT

Appraising the plant, including boilers, settings, chimney, engine foundations, piping, and buildings, at their cost of \$20,000 and charging 5 per cent (a generous allowance for depreciation and interest) gives a yearly charge of \$3,000.

Now from tests made for the purpose it was found that the heating of the mill and process work alone required:

Heating and Mill-Process Work

Day, 178 × 11 = 1,958 boiler horse-power hours
Night, 70 × 13 = 910 " " "

Total heating and process . . . . . . 2,868 boiler horse-power hours

Engine Power

172 i. h. p. × 11 (hours) = ... 1,892 engine horse-power hours
(19.32 ÷ 31.5) × 1,892 = ... 1,162 boiler horse-power hours
2,868 heating alone

Total boiler output = ... 4,030 boiler horse-power hours

Hence the proportion of steam used for power is 1,162 ÷ 4,030 or 28.8 per cent of the steam produced in the boiler plant. Therefore only 28.8 per cent of boiler-plant charges should be made to the engine. The appraisal of the boiler plant, building and stack

is \$13,000 and the engine plant and building \$7,000. Therefore the depreciation and interest chargeable to power production alone is 28.8 per cent of 15 per cent of \$13,000 plus 15 per cent on \$7,000.

15 per cent of \$7,000	\$1,050
28.8 per cent of 15 per cent of \$13,000	562

Total Depreciation and interest to power \$1,612 per year

With 300 working days this would be $1,612 \div 300 = \$5.38$ a day

In the all-day engine test the average indicated horse power developed for 11 hours was 172.8, so that $172.8 \times 11 = 1,901$ indicated horse-power hours were produced.

It is obvious that the depreciation and interest cost per horse power is inversely proportional to the number of horse-power hours produced, so that in taking 172.8 horse power, a low average, the resulting horse-power cost will be on the safe side, that is to say, sufficiently high.

II—The Depreciation and Interest Charge is $\frac{\$5.38}{1901}$ = \$0.00283 per indicated horse-power hour.

The Operating Charges are:

Total operating charges based on actual days operation in year 1908 and also in August, 1909 (the September operating charges were less than August with the same number of operating days), = \$27.60

Now the cost of coal included in "Total Operating Charges" amounts to $4,112 \div 7,158$ or 57.3 per cent of these charges. Hence daily operating charges less fuel are:

100 per cent — 57.3 per cent = 42.7 per cent of \$27.60 .... \$11.78

By analyzing this charge, it is evident that not over half is chargeable to the engine plant.

$11.78 ÷ 2 = $5.89 = daily operating charge to engine power and $5.89 ÷ 1901 =

III—Operating Cost per Indicated Horse-power Hour = $0.003098

Charges per Indicated Horse-Power Hour

I—Fuel cost	$0.002648
II—Depreciation and interest	0.002830
III—Operating (less fuel)	0.003098
IV—Total Cost per Indicated Horse-power Hour	**$0.008576**

Now a kilowatt without consideration of losses is equal to 1⅓ horse power. But in converting from indicated horse power to current at the generator terminals, an allowance for engine friction and dynamo loss makes it necessary to multiply the indicated horse-power cost by 1.54 to get the cost of an effective kilowatt hour.

V—Cost of Effective Kilowatt Hour (1.54 × $0.008576) = $0.0132

COST PER KILOWATT HOUR IF 300 INDICATED HORSE POWER WERE DEVELOPED CONTINUOUSLY ON THE DAY RUN INSTEAD OF 172 INDICATED HORSE POWER AS FOUND

300 indicated horse power × 11 (hours) = 3,300 indicated horse-power hours.

Coal per indicated horse power would remain substantially the same if the engine is speeded up 10 per cent and run at 21 per cent overload to produce 300 horse power. Under these conditions and if the engine is overhauled and tuned up, the same steam consumption per horse power is a fair assumption. Hence:

I—Fuel Cost per Indicated Horse Power per Hour = \$0.002648

Depreciation and Interest. The conditions now would be:

The boiler plant produces

For heating and mill work.....	2,868	boiler horse-power hours
For the engine $11 \times 300 \times (19.32 \div 31.5)$......	2,024	" " "
Total boiler horse-power hours...	4,892	

Engine consumes $2,024 \div 4,892 = 41.4$ per cent of the total boiler output.

Hence depreciation and interest chargeable to engine is:

15 per cent of \$7,000 a year............	\$1,050
41.4 per cent of 15 per cent of \$13,000 a year..............................	807
Total yearly......................	\$1,857

This is $1,857 \div 300 = \$6.19$ a day, and divided by 3,300 horse-power hours gives:

II—Depreciation and Interest per Indicated Horse-power Hour = \$0.001876

The daily operating charge of \$5.89 to engine is the same as when only 172 horse power was developed. Hence for 300 indicated horse power or 3,300 indicated horse-power hours

III—Operating Charge per Indicated Horse-power Hour is \$5.89 ÷ 3,300 = \$0.001785

Summing up items I, II and III, the total cost of producing an indicated horse-power hour would be:

I—Fuel per indicated horse-power hour	$0.002648
II—Depreciation and interest per indicated horse-power hour	0.001876
III—Operating cost per indicated horse-power hour	0.001785
Total cost per indicated horse-power hour	**$0.006309**

The cost per effective kilowatt hour is then (multiply by 1.54) $0.00971 or less than one cent per kilowatt hour.

It is therefore determined that under actual conditions this plant is making its own power, including all charges, for $0.0132 per kilowatt hour as compared with $0.020 to $0.025 per kilowatt hour for purchased electric power.

If the present plant is speeded up 10 per cent and operated at an average of 300 horse power on the day run of 11 hours, a kilowatt hour can be produced for less than one cent as above determined.

The conclusion is that it would be a losing proposition to discontinue the present engine plant.

ENGINE TEST

An *all-day test* was made on the main engine to determine the indicated horse power every 15 minutes and to find the steam and fuel consumption and therefore the fuel cost per horse power per hour under actual operating conditions.

For this purpose, boiler No. 1 was connected and blanked off exclusively to the engine with its condenser and reheater.

Date	Oct. 25, 1909
Duration 7.30 a. m. to 6.00 p. m.	10.5 hours
Engine running	10.417 hours

Steam pressure	118 lb.
Vacuum of condenser	26 inches
Temperature of feed water	161 degrees
Temperature of injection water	54 degrees
Temperature of water leaving condenser	84 degrees
Maximum indicated horse power	218
Minimum indicated horse power (at noon hour)	95
Average indicated horse power	172.8
Rated h. p. of engine at 86 r. p. m.	225
Average indicated horse power divided by rated h. p.	76.8 per cent
Steam used per indicated horse power per hour under factory conditions, including steam to condenser and receiver	19.32 lb.
Total steam used in 10.5 hours	34,795 lb.
Total steam used per hour (10.417)	3,340 lb.

(For cost per indicated horse-power hour see "Cost of producing power with present equipment.")

CAPACITY

In later tests a card showing 265 indicated horse power was taken from the engine. This was at the heaviest part of the day load, when the lighting generator was on. This was the heaviest load obtained, although the indicators were kept on the engine for more than a week. It seems quite possible, however, that the engine may at times carry loads up to 300 horse power as claimed by the engineer, especially when the mill is working up to the full capacity on heavy grades of stock.

This engine may therefore be considered as doing its full duty as regards load.

Electric Power. Cost of Making versus Buying

To take care of full motor load now purchased—day 91 horse power, 68 kilowatts, night 109 horse power, 81.7 kilowatts. It would be well to put in a 90-kilowatt generator set. It would operate at times, for short periods, at as low as $10 \div 90$ = one-ninth capacity but on the average at about $27 \div 90$ = about one-third normal rating. If all the motors which run at one time should happen to carry their full rated load, then the 90-kilowatt generator set would also work at nearly normal full load, and would in addition provide for temporary overloads as high as 25 per cent, i. e., up to 112.5 kilowatts.

The best economy with such an arrangement for displacing the present purchased electric power would be to run this unit non-condensing and utilize the exhaust steam from same in the present heating-fan coils, in the proposed new fan coils, in a feed-water heater, and also for heating the office. In warm weather when the exhaust-steam heating became less than 25 per cent of the steam produced by the engine, the engine should be run condensing.

For purposes of comparison with purchased electric power the most disadvantageous conditions will be assumed, i. e., the exhaust not utilized at all and engine run non-condensing. Further, the engine will be assumed to be a simple Corliss of 125 revolutions.

The average water consumption would be high under these variable conditions of load and will be put at 30 pounds per indicated horse-power hour.

Under these conditions the fuel cost per indicated horse power per hour would be $30 \div 8.03 = 3.736$ pounds of \$2.00 coal. That is $(3.736 \div 2{,}000) \times \$2.00 = \$0.003736$ per horse power per hour; or this equals \$0.00635 for coal per kilowatt hour (at 1.7 indicated horse power = 1 kilowatt at switchboard).

I—Fuel Cost per Kilowatt Hour (1.7 × \$0.003736) non-condensing and with no exhaust utilized = \$0.00635.

The depreciation and interest chargeable against this new unit would be only the charges against the new investment. To be on the safe side call the investment \$8,000 to cover changes in engine room, foundations, piping, engine, generator, exciter, switchboard, wiring and all possible expenses. \$8,000 × 15 per cent = \$1,200 per year ÷ 300 = \$4.00 per day.

Dividing this by the present small load of 582 kilowatt hours per day of 24 hours we shall have as

II—Maximum Depreciation and Interest with *present* load based on an investment of \$8,000, (\$4.00 ÷ 582) = \$0.00688.

Maximum Depreciation and Interest based on probable future load of 1,000 kilowatt hours, (\$4.00 ÷ 1,000) = \$0.00400.

Now the operating cost outside of fuel stated in (I) would not be increased except for oil and waste and half the time of a night man who would attend to the boilers as well. The oil and waste would be more than balanced by the saving in coal by the employment of the night man. Hence the added operating cost properly changeable against the new unit will be:—

III—Operating Cost per Kilowatt Hour based on *present* load (1.165 ÷ 582)	\$0.00201
Operating Cost per Kilowatt Hour based on *probable future* load of 1,000 kilowatt hours per 24 hours (1.165 ÷ 1,000)	\$0.001165

Summary of Costs for Proposed New Unit Per Kilowatt Hour with Present Electric Load

I—*Fuel*, with ½ exhaust utilized the year round, a safe estimate based on tests and data of the investigation—55 per cent of total steam chargeable to engines (0.55 × $0.00635)	$0.00349
II—*Interest, Depreciation, etc.*, based on *present* load of 582 kilowatt hours per 24 hours	0.00688
III—*Operating Charges* based on *present* load—582 kilowatt hours	0.00201
Total per kilowatt hour	$0.01238

Per Kilowatt Hour with Probable Future Load

I—*Fuel* with ½ exhaust utilized, 55 per cent of steam chargeable against engine (0.55 × $0.00625)	$0.00349
II—*Interest, Depreciation, etc.*, based on 1,000 kilowatt hours per 24 hours	0.00400
III—*Operating Charges* based on 1,000 kilowatt hours	0.00116
Total per kilowatt hour	$0.00865

These figures do not include the use of a condenser in the warm months.

Furthermore, as the plant grows the kilowatt-hour cost will decrease below the above figures, and the constantly increasing use for exhaust steam will still further reduce the cost of power from this unit.

Electric Power Required

From the bill for electric power for the year 1908, $3,819.84 ÷ 263 working days = $14.52 a day. At 2½ cents per kilowatt hour, the amount of

power purchased must have been 1,452 ÷ 2.5 = 582 kilowatt hours a day on the average. Based on a 20-hour working day for the motors, this would mean an average continuous power of 582 ÷ 20 = 29.1 kilowatts.

An all-night test of the electric load gave an average load of	21.5	kilowatts
A minimum load of	10	"
A maximum load of	31.3	"

Test made in the day time (nine hours) showed the load to vary from 1.64 at noon and 9 kilowatts at other times to 37 kilowatts. The average for eight hours (excepting noon) was 32.5 kilowatts. Still another test showed as high as 52.5 kilowatts in the day time.

Figuring on the rating of the motors would give possible normal motor loads as follows:

Day Time	*Horse Power*	*Night Time*	*Horse Power*
Machine shop motor	7	Fulling-room motor	75
Scouring motor	35	Heating-fan motor	30
Heating fan	30	Power-pump motor	4
Power-pump fan	4		
Spinning-room fan	15		
	91		109
(or 68.3 kilowatts)		(or 81.7 kilowatts)	

The average electric-motor load may be considered at present as about 30.2 kilowatts.

Day-time test (9 hours) large heating fan not on, average load was	30.2	kilowatts
The possible maximum according to rating of motors on in day time	68.3	"
Large fan motor in test consumed about	13	"
(The belt was slipping so this power is below normal)		

Night average electric load	21.5	kilowatts
The possible maximum according to rating of motors on at night	81.7	"

NOTE: The fulling-room is run by engine in day time and attached to 75 horse-power motor at night.

REPORT ON OPERATION AFTER INSTALLATION OF RECOMMENDATIONS

INDEX TO REPORT

OBJECT OF TESTS

The object of the following tests was to determine the present cost of generating power under the new conditions; and further to compare the present costs with those obtained when the plant was tested in October and November, 1910, following which test recommendations were made covering certain changes which are now in operation.

The principal changes that were recommended and adopted were as follows:—

1—Install a 120 horse-power simple Corliss engine with generator to displace about 30 kilowatts of purchased electric power and to provide 60 kilowatts for taking care of increases in machine load as added from time to time.

2—Install a first-class vacuum system and utilize the exhaust steam to heat the mill instead of live steam then in use.
3—Speed up the main engine in order to increase its capacity.

CONDENSED RESULTS OF THE INVESTIGATION

Present Cost of Power.—Under present conditions the total cost of making power, including all charges, with the main engine is $0.01043 per kilowatt hour delivered on the bus bars.

The result of installing the new Corliss engine with a 90-kilowatt generator has been to displace electric power formerly purchased at the rate of $0.025 per kilowatt hour and to make this power a tthe rate of $0.00937 per kilowatt hour when the output is 1,000 kilowatt hours per day, the present low average.

From tests that have been made it was found that over one-half the exhaust steam produced by this new unit is utilized in place of live steam formerly used, based on a year-round consumption. But in the above cost deduction only one-half the heat in the exhaust has been credited to the engine.

It is of special note that the above cost is being produced under disadvantageous conditions. That is to say, the new engine was designed to run on 120 pounds boiler pressure. As a matter of fact, the test was made with only 74 pounds steam pressure at the engine. It is my opinion that a reduction in steam consumption of the engine of 10 per cent would be made by operating with 120 pounds steam pressure instead of 74 pounds as at present, now that the load has sufficiently increased to prevent valve and regulation difficulties.

It will be further noted that the above costs per kilowatt closely agree with those predicted in the writer's report made in the fall of 1909.

The X heating system with Y valves has given satisfaction according to report of the superintendent and engineer, and has resulted in the advantage of being able to heat the buildings with exhaust steam where live steam was formerly and exclusively used. It has also been possible by means of the vacuum system to apply exhaust steam to other purposes where live steam was formerly employed. With the use of the X system no back pressure is required in order to provide circulation of steam in the heating system.

Both power and heating requirements have been largely increased since the writer's visit in 1910. The power comparison may be stated as follows on a basis of 24-hours output:

	Kilowatt Hours
Main engine, 1910	1,150
Main engine, 1912	1,600
Increase of power on main engine	450 or 39 per cent
Old Albany Southern power 1910	666
New engine power displacing above, 1912	1,070
Increase	404 or 61 per cent

Increase of power based on totals:

Former power	1,816
Present power	2,670
Increased power	854 or 47 per cent

As a practical check on the above results the following figures from the Company's books may be quoted: With the 47 per cent increase of power as above shown, the heating requirements of the mill have also been increased at least 50 per cent owing to the large additions that have been made, which additions are very largely exposed to the

weather and contain a very great percentage of window space.

Day Cost.—As a check on statements otherwise derived the following may be quoted: The total day operating cost of power, heat and light for 1910 was $25.40. For the seven months' operation of the new system with about 50 per cent increase of power, heat and light, the day cost was $30.60, showing an increase of $5.20 a day, which is equal to 20.5 per cent increase of operating cost.

Savings.—The work of the new engine and generator from a recent reading is 1,070 kilowatt hours per day, and this load is continually increasing. At the old rate of 2½ cents this would cost $26.75 per day.

The present cost with new engine for the same power as above is 1,070 × $0.00937 = $70 per day, thus making a saving of $16.75 per day or $5,025 per year. This is a conservative figure, since the power requirements are increasing steadily with the installation of new machinery, and the use of exhaust steam is also growing.

If you had accepted the offer of the electric power company to supply *all* your power at the rate of 2 cents per kilowatt hour instead of adopting the plan recommended in the first report, your total power expense would be $8,010 a year more than it is at present. That is to say, you would be paying for 2,670 kilowatt hours per day at a 2-cent instead of a 1-cent rate.

A larger operating charge than is truly proportional has been made against the new engine in order to be on the safe side in the estimates of savings accomplished. Otherwise the predicted and actual costs for power under the new system will be seen to agree closely by comparing the original report with the present one.

The saving due to speeding up the main engine

may be considered as equal to the extra power so provided compared to purchasing this power. On the basis of a safe estimate, as elsewhere calculated in this report, this saving amounts to at least $450 per year.

The total savings from these two changes are therefore $5,025 + $450 = $5,475 per year. If $150 per year be deducted as interest charges on changes to heating system, the net saving is $5,325 per annum. The net interest or dividend on investment is 5,325 ÷ (7,000 + 1,000) = 66.5 per cent.

PLAN OF TESTS

TEST No. 1.—The fuel cost of the main engine per horse power and per kilowatt hour was obtained in a 4½-hour run by measuring the water fed to the single boiler which was connected to supply steam to this engine alone. The water was measured in barrels calibrated by weighing. Indicator cards were taken periodically throughout the test together with switchboard readings showing electrical output at the bus bars.

TEST No. 2.—The fuel costs of the new simple non-condensing engine were obtained in a separate test of three hours' duration in the same way as in Test No. 1.

TEST No. 3.—The total steam produced by both boilers under present conditions was obtained by making a capacity test on the two boilers at the same time, all water fed being measured in the calibrated barrels and all necessary readings being taken.

During this test a record was kept of the electrical output of each engine, from which figures, and from the steam per kilowatt hour determined in Tests Nos. 1 and 2, it was possible to learn what proportion of the total steam generated in the boilers is consumed by the engines and by the mill respectively under working conditions in cold weather.

Electrical Testing.—In addition to the above work the electrical installation was inspected and the meters were tested for accuracy. Following this, a report was made on electrical conditions as found, together with certain recommendations for improvement.

NEW ENGINE—STEAM PRESSURE

Owing to the light fractional load at one time carried by the new engine it was necessary to install a pressure-reducing valve. The day load has now increased to a point where this reduced pressure (75 pounds) is unnecessary and furthermore is wasteful of steam.

I would therefore recommend by-passing enough steam from the mill line, by means of the present by-pass, to build up the pressure in the engine line to 115 pounds. This will increase the horse power of the engine more than 50 per cent, give it a much earlier cut-off, and make a material reduction in the consumption of steam.

At night or at times when there may be too light a load, the by-pass valve can be closed and the engine operated through the reducing valve.

EFFECT OF SPEEDING UP MAIN ENGINE

The effect of speeding up the main engine may be stated in terms of dollars as follows:

Present output of main engine, per day	1,600 kilowatt hours
Former output 173 horse power equal to, per day	1,150 kilowatt hours
Per cent increase in load	39 per cent

Although a part of this 39 per cent could have been added to the former load, it would not have been good or economical practice to have added more than 30

per cent at the former speed. Hence, a mimimum estimate of the power gained by speeding up may be considered as equal to about 10 kilowatt or 100 kilowatt hours per day which would have otherwise to be purchased at $2\frac{1}{2}$ cents or made by the new engine. But considering this as a separate proposition, the saving can be safely estimated at 1.5 cents per kilowatt hour, or \$1.50 per day, or \$450 per year.

Test No. 1. Main Engine and Generator

Engine: Knowlson & Kelly $14\frac{1}{8} \times 26\frac{1}{8} \times 36$, 97 to 100 r. p. m., cross-compound condensing, belt-connected to a 180-kilowatt alternating-current generator, 240-volt, 3-phase, 25-cycle

Date of test	Jan. 20, 1912
Duration of test	$4\frac{1}{2}$ hours
Boiler pressure by gage, average	109.4 lb.
Vacuum by gage, average	25.7 inches
Receiver pressure (variable)	9 to 20 lb.
Temperature of feed water (for test only)	89 degrees
Maximum kilowatt reading on bus bars	180
Minimum kilowatt reading on bus bars	125
Average kilowatts per hour	155
Average indicated horse power per hour	232.5
Actual weight of steam per kilowatt hour	29.14 lb.
Actual weight of steam per indicated horse-power hour	19.43 lb.
Indicated horse power per kilowatt on bus bars	1.5
Rated capacity of engine at 100 r. p. m.	300
Percentage of rated capacity developed ($232.5 \div 300$)	77.5 per cent

Coal per kilowatt hour on bus bars..	3.63 lb.
Coal per indicated horse power per hour..........................	2.42 lb.
Fuel cost per kilowatt hour........	$0.00363
Fuel cost per indicated horse-power hour..........................	$0.00242

Note: The fuel cost of making steam in the boilers is taken from tests made in October and November, 1910. This is done as no change which would affect this cost has been made in the boiler plant.

DEPRECIATION AND INTEREST CHARGES

No increase of these charges has occurred since Report of 1910, and referring to that report, it will be seen that the total amount was $3,000 per year on the whole steam plant.

Now from data obtained it is determined that 70 per cent of the maximum boiler output is required for purposes other than the main engine. (See former report.) A greater proportion of steam is now used for heating.

Depreciation and interest of boiler plant chargeable to main engine per year is 30 per cent × (15 per cent × 13,000) = ..	$585.00
Depreciation and interest on engine plant, 15 per cent × 7,000..................	1,050.00
Total depreciation and interest charged to main engine per year................	$1,635.00
Total depreciation and interest charged to main engine per day..................	$5.46

Taking the average output of the main engine set at 1,600 kilowatt hours we have:

Depreciation and interest charge per kilowatt hour on main engine set ($5.46 ÷ 1,600)..............................	$0.00341

OPERATING CHARGES

Total operating charges based on actual days operation from June 1 to Dec. 31, 1911, i. e., 7 months of average temperature, during which time the new power system was in operation (fuel expense deducted) per day	$12.02
By analyzing this item, it is found that	
Expense chargeable to engine room (both engines) is, per day	8.00
Operating charge to new engine per day	2.58
Operating charge to main engine per day	5.42
Operating charge to main engine per kilowatt hour, 5.42 ÷ 1,600	$0.00339

Collecting the three cost items before determined, we have:

PER KILOWATT HOUR

1—Fuel cost per kilowatt hour	$0.00363
2—Depreciation and interest charges	0.00341
3—Operating charge except fuel	0.00339
Total cost per kilowatt hour	$0.01043
Total cost per indicated horse-power hour	$0.00696

Test No. 2. New Engine and Generator

Engine: Knowlson & Kelly 14 × 30, 126 r. p. m. (during test); simple Corliss, non-condensing, belt-connected to a 90-kilowatt alternating-current generator, 240-volt, 3-phase, 60-cycle

Date of test	Jan. 20, 1912
Duration of test	3 hours
Steam pressure low side of reducing valve	74 lb.
Temperature of feed water (for test only)	53 degrees

Maximum kilowatt reading on bus bars	64.4
Minimum kilowatt reading on bus bars	46.0
Average kilowatts per hour	60.3
Indicated horse power per kilowatt	1.6
Average indicated horse power per hour	96.6
Actual weight of steam per kilowatt hour	48.1 lb.
Actual weight of steam per horse-power hour	30.0 lb.
Rated capacity[1] of engine at 75 lb. initial pressure and 126 r. p. m. at 25 per cent cut-off	100 horse-power
Percentage of this rated capacity developed	96.6 per cent
Coal per kilowatt hour	5.98 lb.
Coal per indicated horse-power hour	3.74 lb.
Fuel cost per kilowatt hour	$0.00598
Fuel cost per indicated horse-power hour	$0.00374

Note: The cost of making 1,000 pounds of steam is taken from boiler tests made in October and November, 1910.

DEPRECIATION AND INTEREST CHARGES

There is no charge to these accounts from the boiler house against this new engine; but the cost of the new engine, generator, foundations and its part of the switchboard and wiring are direct charges against the power developed.

The cost of these items is taken from data on the books of the mill and amounts to less than $7,000. This cost is made up as follows:

[1] At 115 lb. initial pressure the rating of this engine would be 170 horse power.

Engine	$1,750
Generators	1,700
Foundations	500
Piping	300
Wiring and construction	2,000
Regulators	205
Generator switchboard	400
Total cost of new engine and generator	$6,855
Call this for safety	$7,000

Depreciation and interest charges per year, 15 per cent of $7,000	1,050.00
Depreciation and interest charges per day	3.50
Depreciation and interest charges per kilowatt hour—($3.50 ÷ 1,000 kilowatt hours per day)	0.0035
Other charges against this generating set are:	
Cost per day for additional wages (from the books)	1.58
Oil, waste and supplies (a very high figure from the books, which can be reduced)	1.00
Total other charges per day	2.58
Total other charges per kilowatt hour ($2.58 ÷ 1,000 kilowatt hours[1])	0.00258

COST PER KILOWATT HOUR

Summing up the Above Cost Items

Depreciation and interest charges	$0.00350
"Other charges"	0.00258
Fuel cost	0.00598
Total cost per kilowatt hour	$0.01206
Total cost per equivalent indicated horse-power hour	$0.00754

[1] This total of 1,000 kilowatt hours per day was obtained from monthly readings of the individual integrating watt-meter attached to generator of this engine.

In computing the above power cost no credit has been allowed for the utilization of exhaust steam. When all the exhaust is utilized, as in cold weather, it takes the place of 90 per cent of its weight of live steam. If one-half of the exhaust produced by this engine is utilized on an average the year round (a safe estimate) the fuel cost of steam chargeable to power will be reduced 45 per cent and the total cost per kilowatt hour will be as follows:

Cost Per Kilowatt Hour

Depreciation and interest	$0.00350
"Other charges"	0.00258
Fuel cost	0.00329
Total cost per kilowatt hour	$0.00937

If it be assumed that all the exhaust of this engine is wasted all the year round, the following comparisons can be obtained:

	Exhaust of New Engine all Wasted	One-half Exhaust Utilized
1,000 kilowatt hours per day, present work of new engine	$12.06	$9.37
1,000 kilowatt hours at 2½ cents with purchased power	25.00	25.00
Saving per day by new engine	12.94	15.63
Saving per year by new engine—300 days	3,882.00	4,689.00
Interest (profits) on investment	55.4 per cent	67.1 per cent

COST DATA FROM THE BOOKS

Average cost per day for Power, Heat and Light.—These figures include everything but interest charges.

	Aug.	Sept.	Oct.	Nov.
Old system, purchasing power...	$35.25	$37.13	$31.89	$37.36, 1910
New system, making own power..	$28.96	$26.57	$29.40	$33.51, 1911

Average day cost 1908	—$27.60—	operating days	263
" " " 1909	— 25.40—	" "	284
" " " 1910	— 31.65—	" "	303
" " " 1911	— 30.60—	" "	178
		(May 31–Dec. 31)	

The new system was started in May, 1911, so that up to time of second investigation it has been running about eight months. The average day cost for 1911 is taken from May 31 to Dec. 31, giving seven full months of about average temperature. About $1.00 per day should be added to these figures, which is paid to maintain the contract with the power company.

The average daily power output for 1911 as compared with Oct. 25, 1909, is given opposite.

"Day cost" does not include interest charges, but covers power, heat and light.

The table on page 559 gives a rough comparison showing that while the day cost increased 20 per cent, the power used increased 33½ per cent. These figures are favorable, especially in view of the large additions to the mill which have greatly increased the heating requirements.

Power developed or used per day, 24 hours	Horse-power hours, 1909	Horse-power hours, 1911
Main engine 173 x 11 . .	1,903	2,190
Purchased power 551 kilowatt hours or new engine and generator .	735	1,332
Total per day	2,638	3,522
Average day cost	$25.40	$30.60

Notes on Charges Against Engines

The following principles were employed in computing the cost per kilowatt hour on the two generating sets.

The analysis of these percentage charges is as follows:

A. Only about 30 per cent of the maximum capacity of the boilers is required for the main engine, and while the remaining 70 per cent is not in use for mill and heating purposes continuously, yet this part has to be held in reserve strictly for these purposes. Therefore about 30 per cent of the fixed charges on the boiler plant is chargeable to the main engine and to the power it produces.

As the exhaust steam from the new engine is utilized for heating, thus displacing the use of an equivalent amount of live steam, no additional boiler capacity was required on its account and therefore no fixed boiler-plant charges can be made against it.

B. In the case of either engine, it is evident that their necessary fixed charges are to be charged against their respective power outputs.

In this connection it will be understood that the electrification of the mill and of the main engine constitute charges against the mill, and not against the cost of producing power. This is evident, for electric transmission was decided upon because of its benefits throughout the mill in various ways, but not at all for the purpose of producing power or for reducing its cost. It must properly be considered as a system of transmitting the energy that was already being produced in a more convenient manner, the mill and not the engine room being considered.

In the case of the new engine, on the other hand, the fixed charges resulting from the installation of its generator and switchboard must be included in the determination of the cost of power developed by this engine. This is evident for the reason that no other kind of an installation would have performed the function required; that is, of replacing the purchased electric power. This was the purpose of the new engine, and a generator and its fixings formed part of the necessary equipment.

C. No operating charges on the boiler plant except fuel are charged against the new engine, since these costs are in no way changed, and since the combined night engineer and fireman is charged half-and-half to the engine and boilers.

The additional wages of the day engineer are charged against the new engine, and one-half of the wages of the night engineer and fireman.